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U. S. GUNBOATS *DUBUQUE* AND *PADUCAH.*

BY COMMANDER W. N. LITTLE, U. S. NAVY, MEMBER.

DESCRIPTION.

An Act of Congress making appropriation for the naval service, approved July 1, 1902, authorized the construction of two 12-knot gunboats of about 1,000 tons each, all parts of which were to be of domestic manufacture.

Bids for the construction of both vessels were duly advertised, and the contracts for their construction awarded separately to the Gas Engine and Power Company and Charles L. Seabury & Company, Consolidated, of Morris Heights, New York.

The contract for gunboat *No. 17*, the *Dubuque*, is under date of May 29, 1903. For gunboat *No. 18*, the *Paducah*, the sister ship, the contract was not signed until July 6, 1903, after re-advertisement.

The contract price for the *Dubuque* was two hundred and ninety-five thousand dollars. For the *Paducah* it was three hundred and fifty-five thousand dollars.

This price was for hull, machinery and equipment, excepting stores, and provided for the installation of the armament

and ordnance outfit, complete, by the contractors after it is supplied by the Government.

The contract time for completion of the *Dubuque* was twenty months; afterwards extended, because of difficulty in obtaining materials and because of strikes and labor conditions in vicinity of works, six months, to June 1, 1905.

For the *Paducah* the contract time for completion, twenty months, was extended four months, to July 6, 1905.

As these vessels are, one built and the other building, from the same plans and specifications, prepared by the Navy Department, a description of one should suffice for the other.

The *Dubuque* is a composite-built gunboat of 1,080 tons displacement, with two masts, two funnels, flush main and gun decks and an interrupted berth deck. The principal dimensions and characteristics of the hull are as follows:

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet	174
on L.W.L., feet	174
over all, feet and inches	200- 9
Beam, extreme, feet	35
at L.W.L., feet	35
Ratio of length to beam	4.971 to 1
Depth, molded, main deck, feet and inches	22- 9⅝
Draught, normal, mean, feet and inches	12- 3
mean on trial, feet and inches	12- 3
Displacement, normal draught, tons	1,085.3
trial, tons	1,085.3
per inch at L.W.L., normal draught, tons	10.51
Area midship section to L.W.L., normal draught, square feet	348
L.W.L. plane (draught 12 feet 3 inches), square feet	4,470.7
Wetted surface, square feet	6,900
Center of gravity of L.W.L. plane abaft M.S., inches	3
buoyancy above bottom keel, feet and inches	7- 3
forward of M.S., feet and inches	2-11¼
Transverse metacenter above center of buoyancy, feet and inches	9- ⅞
Longitudinal metacenter above center of buoyancy, feet	194
Coefficient of fineness, block	.509
of M.S.	.812
L.W.L.	.732
Cylindrical coefficients, longitudinal	.627
vertical	.904
Number of frames	116
Number of compartments	51

ARMAMENT.

Six rapid-fire 4-inch guns; four rapid-fire 6-pounders; two rapid-fire 1-pounders; two Colt automatic.

The ship is propelled by twin outward-turning screw propellers driven by a pair of three-cylinder triple-expansion engines. Steam is generated by two Babcock & Wilcox marine boilers under a gauge pressure of two hundred and sixty-five pounds per square inch, intended to be reduced to two hundred and fifty pounds (maximum) at the engine.

No armor or protective decks are present.

HULL.

The above water form of the hull is unique in appearance for a naval vessel although not uncommon in a steam pilot boat or light ship. The stem slopes well aft below the water line and the heel is reduced or cut away for ten feet from stern post to facilitate turning and to mimimize the tactical diameter.

The flaring bow affords excellent deck space for working the anchor gear and should be advantageous in a head sea.

The rudder, which is of bronze, of the barn-door type, is suspended by three pintals on its forward edge and its lower edge is thirty-three inches above the lowest part of the keel and is about in line with the heel. Its area is fifty-two square feet.

The frames of the hull above the floors are fifteen-pound channel bars, spaced twenty inches apart, excepting near the bow and the stern.

The vertical keel is thirty inches deep between inner and outer bottoms, lightened by oval holes. The depth is uniform between frames 28 and 100, decreasing forward. Under the engine room, seven feet each side of the keel plate, are longitudinals between inner and outer bottoms. On the starboard side the longitudinal girder merges into the bunker bulkhead. On the port side it ceases at the fireroom; the port bunker (fireroom) bulkheads extending to the outer bottom. Additional intercostal girders support the inboard

seatings for the engine bed plate. Below and for twenty-six inches above the water line the hull is sheathed and coppered. Excepting for a small space above the bilges the hull is plated overall, the plating being about fourteen pounds to the foot. Under the boilers and magazine spaces are water bottom compartments, seven in all, having a capacity for forty-eight and eight-tenths (48.8) tons of fresh water. Four of the compartments under the boiler compartment are fitted for reserve tanks, and have an average capacity each of four and thirty-five-hundredths (4.35) tons of fresh water. These reserve tanks can be pumped from one to another. There is a trimming tank forward, of 525 cubic feet capacity and one aft of 214 cubic feet which can be pumped through the secondary drain, and filled from the sea or through hose and manhole at top.

The planking is from four and one-half to three-inch yellow pine.

The berth and gun decks are steel throughout, covered with linoleum. The main deck is stiffened by stringers, waterways and plating in the waist and planked overall.

On the main deck are the pilot and chart house, over which is the bridge, equipped as customary with electric and mechanical annunciators to engine room and an electric revolution indicator from each engine shaft of the vibrating-arrow type. Abaft and contiguous with the pilot and chart house is the galley, followed by the boiler-room hatch and two funnels, an athwartship passage, a wireless-telegraphy house, the engine-room hatch, an athwartship passage, hatch to berth deck, ward-room skylight, standard compass platform, main mast, hatch to "cabin country," and, finally, the hand-steering gear.

The gun-deck space is divided and occupied as follows, the spaces considered in order from forward:

A paint room in the eyes of the ship contains numerous tanks (nine), with glass gauges, for oil; others (five) for mixed paints and dry materials. It is lighted, as is the gun deck, throughout by air ports and dead lights of ample size, and is well fitted with shelves. Abaft the paint room is a midship passage flanked by the crew's water closet on the starboard

side and the crew's wash room on the port side. In the passage are valves controlling the head-flushing system, three spigots for salt water from same pipes and a locked spigot for fresh water. The crew's lavatory contains seven tilting and removable glazed basins, each under a spigot and over a trough, and four shower compartments, with hot-water facilities and a spigot for bucket filling.

In the next compartment, the second from forward, are the steam capstan engine, worm wheel and vertical shaft, and two chain compressors of the ordinary type; on the starboard side, the ship's prison and scuttle butt; on the port side, a storeroom and a large ice chest; next are two 4-inch mounts and ports, one each side. Under the pilot house is the armory.

In the third compartment are, amidships, the firemen's washroom, with six basins and spigots, two shower baths, and a fresh water tank of 300 gallons capacity, tiled floor, two mounts each side for 6-pounders, the fireroom hatch way and drum rooms, in the wake of which, amidship, is the machine shop, containing a lathe, shaper, drill press and a tool grinder driven by an electric motor; also a small bench with two vises.

Abaft the machine shop is the engine-room hatch trunk, extending, as do all principal bulkheads, from ship's bottom to the main deck, abaft of which is the main hatch to main and berth decks. Abreast those hatches are the following rooms: starboard side—pay office, executive officer's office, executive officer's room; port side—engineer's office, ward room officer's lavatory, bath room and water closet.

Next occurs bulkhead No. 82, abaft of which is the ward-room officers' mess-room amidship and on port side, pantry amidships, flanked and succeeded by eight state rooms and a passage. The captain's cabin, aft of all, extends from side to side abaft bulkhead No. 105, and is about 15 feet wide by 13 feet long. Off and forward of the cabin are one state-room on starboard side and cabin pantry on port side. In the overhanging stern and over tiller room is the captain's lavatory, with bath and water closet of the yacht type, with

tiled floor. The 'tween-deck distance is six feet three inches in the clear.

The berth deck is interrupted by the machinery spaces with neither bulkhead doors nor passages to connect fore and aft.

In the bows over the trimming tank are the sail room and equipment store room to frame 17; crews' space to frame 45, with an intermediate watertight bulkhead at frame 27. In the forward of these two berth-deck crew spaces, *i. e.*, compartment A–11, are located three 120-fathom hawser reels, leads to chain lockers and hatches to store room below. In the second crew space, compartment A–12, are the hatches to fore hold and forward magazine, the steering-engine and dynamo-engine room.

After berth-deck spaces from engine room are the dispensary, sick bay and petty officers' and marines' quarters; abaft of which space, with a watertight bulkhead and door between, is the mess attendants' quarters, three store rooms and tiller room. The absence of a steering engine and dynamo engine from aft affords unusual space for crew and stores in this part of the ship. The tiller arc or quadrant fits closely under the gun deck, occupying very little space.

The sick bay has six beds of the folding or close-stowing type, no woodwork in vicinity excepting oak desks and drawers in dispensary. All light bulkheads are of corrugated galvanized iron. A ventilating electric blower of 2-horse-power is located in a recess on starboard side of after berth-deck passage, *i. e.*, in the mess attendants' quarters.

Below the berth deck, forward of the machinery spaces, occur the trimming tank, dry provision, general and construction store rooms and the forehold (with cable tier) as far as frame 33. In this hold outboard, forward of and adjacent the forward magazine, are the two main fresh-water tanks. Between these tanks in separate compartments are the electrical and ordnance stores. The forward magazine is situated between bulkheads at frames 36 and 45. The 6 and 1-pounders are stowed outboard and the 4-inch amidship. Between the for-

ward magazine and boiler space the bulkhead is sheathed with asbestos millboard and an air space is provided. A similar provision is made for the protection and ventilation of the forward bulkhead of the after magazine in wake of the engine room.

Below the berth deck abaft the machinery spaces, on the port side of the shaft alley, are the paymaster's small stores and petty officers' stores. On the starboard side, and entered from the starboard-shaft alley, are the machinery stores. In the wake of these store rooms are the small-arms and 4-inch magazine and handling room as far as frame 92, abaft of which are the navigator's stores, flanked by hospital stores on port side and marines stores on starboard side, to frame 98. Paymaster's stores occupy the remaining space aft to the trimming tank.

BOATS.

The boats carried are:

	Capacity, men.	Pounds water and provisions.
One 28-foot steam cutter,	35	350
One 28-foot cutter,	40	400
Two 26-foot cutters, each	30	300
One 28-foot whaleboat,	23	230
One 28-foot gig whaleboat,	16	160
One 18-foot dinghy,	10	100
Two 12-foot punts.		

COMPLEMENT.

Commander,	1
Wardroom officers (staterooms),	8
Engine-room force,	29
Special branch,	11
Seaman branch,	73
Artificers' branch,	10
Messmen,	14
Marines,	12
Total,	158

The *drainage system* includes a 7½-inch galvanized, lap-welded steel main drain ¼-inch thick, with terminal suctions in after end of fireroom and engine room, amidships of, and a secondary drain of 4-inch galvanized lap-welded steel pipe $\frac{3}{16}$-inch thickness extending from forward end of the boiler compartment to the after engine-room bulkhead. Secondary drain manifolds are located—one (No. 1) in starboard passage at forward end of boiler compartment, with leads to fore holds and chain lockers, and one at the after engine-room bulkhead nearly amidships, with leads to the shaft alleys and afterbody bilges. Valves controlling the suction forward and aft of the machinery spaces are of the stop-check type, and operated on berth deck, except in shaft alleys, which are operative at the valve only by handwheel below floor plates in engine room. The main drain is connected by a 5½-inch galvanized steel pipe, with both main circulating-pump suction pipes. This connection is made operative by first closing the main injection valve, which act releases and permits the opening of the bilge suction valve at the circulating pump in question, and then opening the 7½-inch special stop check valves operated on gun deck or above floor plates near after bulkhead amidships, one in engine room and one in boiler compartment. There are no Macomb strainers on the main drain. Macomb strainers and 4-inch valves in the secondary drainage system in machinery spaces are three in number, located at the bilge wells, one amidships at the after end of engine room and one about 5 feet to starboard of amidship line between after boiler and bulkhead, one 3-inch branch suction valve and strainer at forward end of engine room. The engine crank pits are drained by connections with the independent fire and bilge pump in engine room and by the main bilge pumps attached to engines.

The double bottoms connected with the drainage system are A–99 and B–95. Compartment B–95 only can be flooded from the sea by a connection at sea suction valve in fireroom. The main drain is pumped only by the main circulating pumps. The secondary drain is pumped by attached engine bilge pumps,

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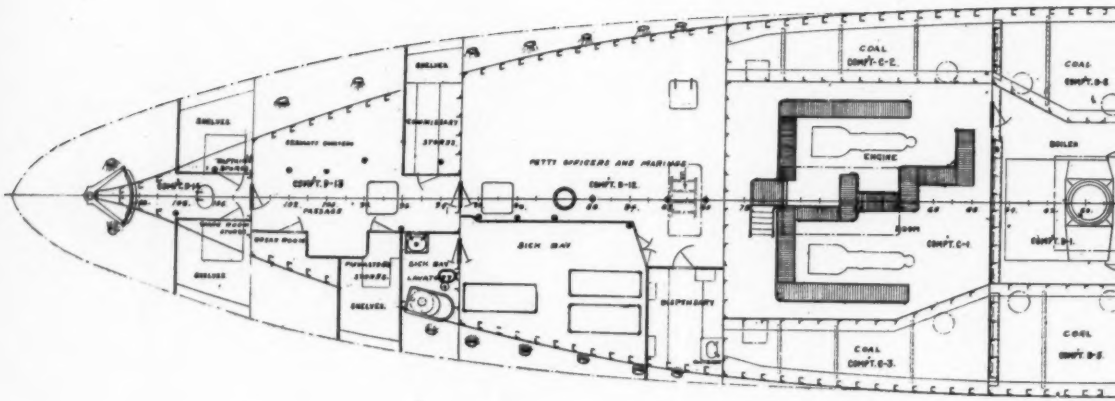


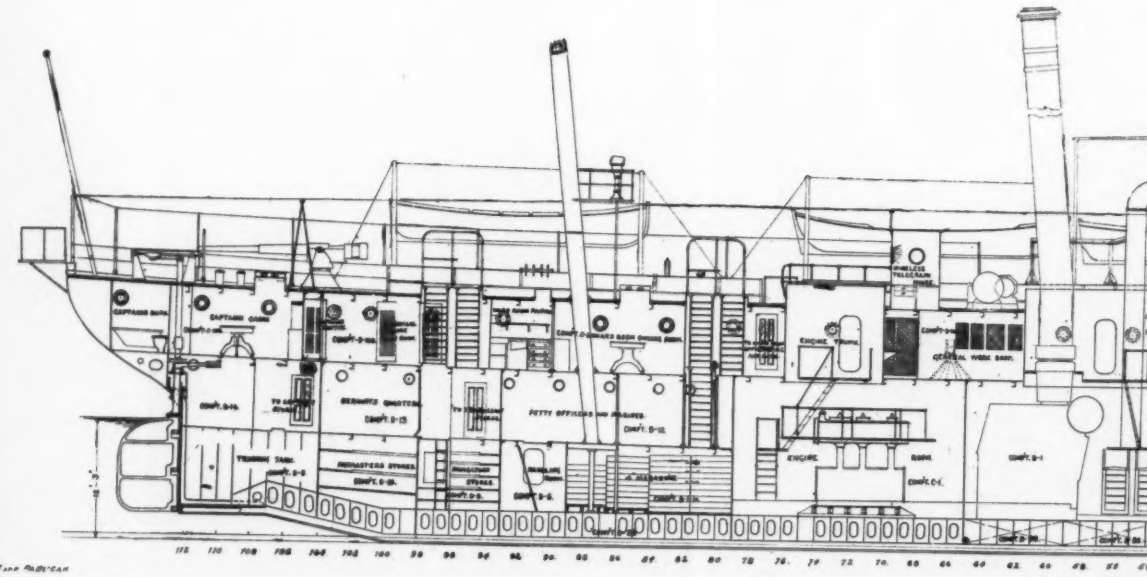
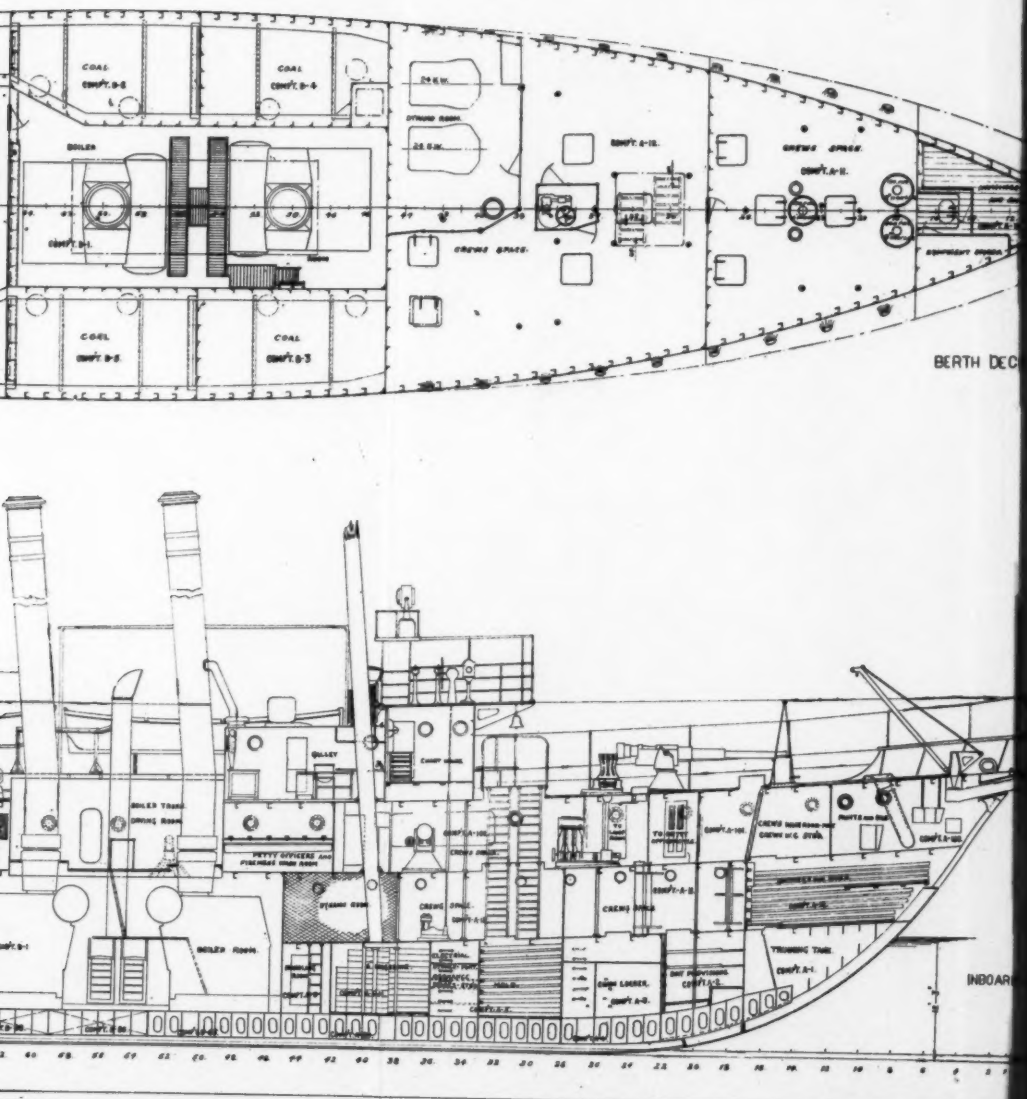


Plate III.—Berth-Deck Plan and Inboard



ND INBOARD PROFILE, U. S. S. "DUBUQUE."

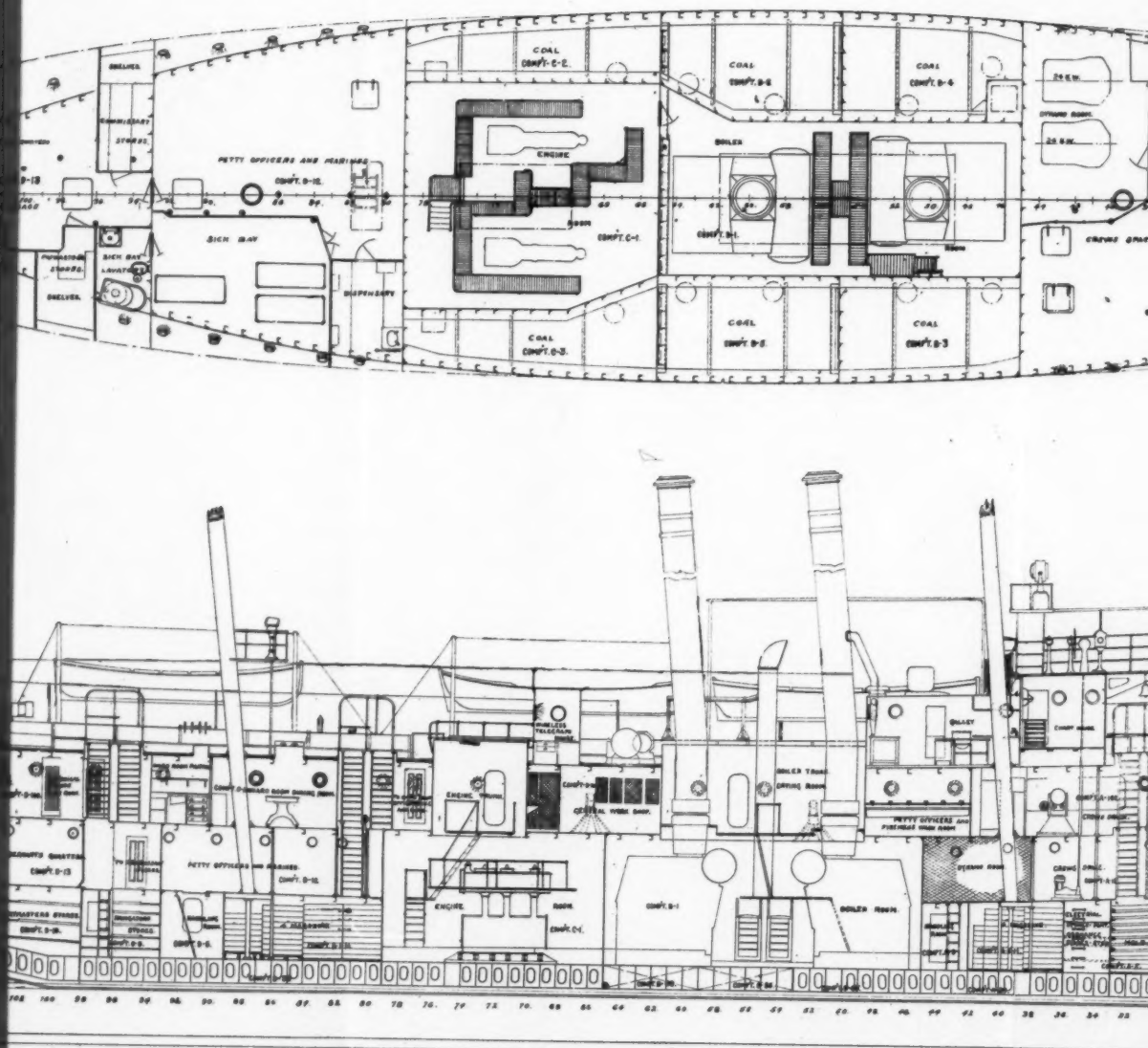
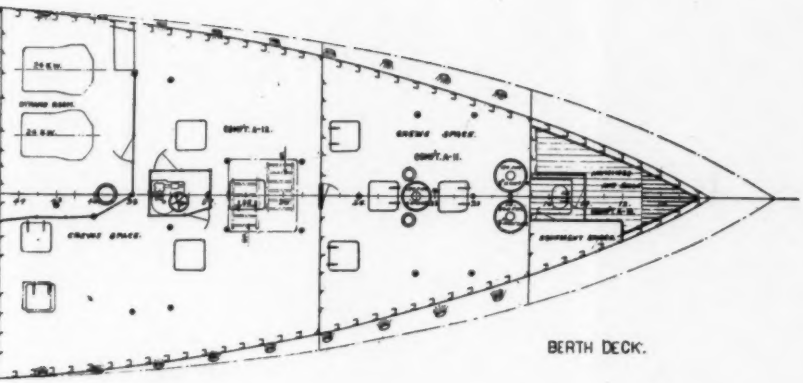
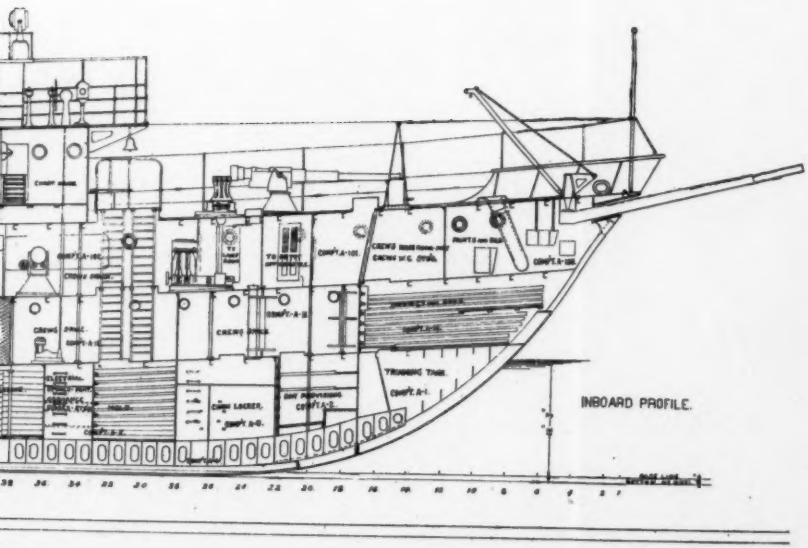


Plate III.—Berth-Deck Plan and Inboard Profile, U. S. S. "Dubuque."



BERTH DECK.



INBOARD PROFILE.

UBUQUE."

by the fire and bilge pump starboard side of engine room, by the auxiliary, feed, fire and bilge pump in fireroom, and by two hand pumps on gun deck.

Manifolds on the drainage system are as follows: No. 1. Forward end of boiler compartment, starboard side; No. 2. At auxiliary feed, fire and bilge pump in fireroom; No. 3. Fire and bilge pump in engine room; No. 4. At after bulkhead in engine room.

FIRE-MAIN SYSTEM.

The 4-inch copper fire main starts from the riser at the fire and bilge pump, in engine room and the forward lead passes through the boiler room on the starboard side under the gun-deck beams.

There is a 2½-inch riser from the auxiliary feed pump in boiler room to the fire main. By closing the gate valve in the main near this connection, the fire main is divided into two systems, one forward and one aft. The "auxiliary feed pump" is connected with the former and the "fire and bilge pump" with the latter.

The distiller circulating pump may also be used on the fire main, through a connection between frames 67–68, in engine room.

There is a connection to the sanitary system between frames 19–20, gun deck, forward.

A relief valve, set at 100 pounds, between frames 29–30, discharges into main deck scupper on starboard side of berth deck.

Cut-out valves are situated at the foot of each riser. Connections for flushing ash chutes are taken from this main.

FRESH-WATER SYSTEM.

The fresh-water tanks in hold may be supplied from fresh-water boat by connections on ship's sides between frames 28–29, port and starboard. From these tanks the reservoir tank on top of galley house can be filled by means of hand pump on forward berth deck.

The supply from distillers may discharge into tank in firemen's wash room and ship's tanks.

The main pierces the gun deck between frames 42–44 from below, leading along under the main deck beams to 80 and 81, at which point it drops to berth deck with branches to pantries and bath rooms.

THE FLUSHING SYSTEM.

The flushing main is connected to the sanitary pump on port main engine. There is also a connection to the distiller circulating pump between frames 69–70 in engine room.

The forward lead passes through the boiler room on the port side, and, continuing along the berth deck, pierces the main deck between frames 20–21, to supply crew's head.

The after lead extends under the gun deck beams, with branches to pantries and bath rooms.

A relief valve, set at 15 pounds, is located in crew's water closet, with discharge into trough.

The flushing water may also be taken from the salt-water discharge from the distillers.

THE VENTILATING SYSTEM.

The ventilating system provides artificial ventilation for all living spaces below the gun deck, including engine room, store rooms, magazine passages, also the air-casing spaces between gun deck and boilers, and between engine and boiler compartments and the magazine adjacent to each; also to the water closets and toilets on the gun deck. As this class of vessels has a high freeboard and hatches opening to the main deck in vertical lines, an exhaust in addition to a supply system was not, excepting for the compartment in which the dynamos are located and the bunkers, deemed necessary. There are six electrically-driven centrifugal fans arranged in as many units or sections of piping. None of the principal bulkheads are pierced by the air ducts excepting at No. 45, where there is a butterfly valve, and the duct supplying air to the double space under gun deck over boilers and to bunkers is of heavy iron pipe. Air ducts connect the bunkers with

the annular space around uptake and smoke pipes without gates or valves.

The following is a list of the fans by their number, designation and the purpose of each:

Fan No. 1. Exhausts from water closets and wash rooms forward of bulkhead No. 23 on gun deck.

Fan No. 2. Exhausts from officers' water closet and bath rooms on gun deck and discharges into engine-room hatch.

Fan No. 3. Exhausts from dynamo-room compartment and discharges into air space over boilers.

Fan No. 4. Supplies fresh air to dynamo room and to all compartments forward of bulkhead No. 45 and below gun deck.

Fan No. 5. Supplies fresh air to engine room.

Fan No. 6. Supplies fresh air to all compartments abaft of bulkhead 79 and below gun deck.

PARTICULARS OF EACH VENTILATING FAN.

Fan No.	Supply or exhaust.	Size.	Rated power.	Cu. ft. air delivered per minute.	Kilowatts absorbed.
1.	Exhaust.	No. 2 Monogram.	½ H.P.	570	.51
2.	Exhaust.	No. 2 Monogram.	½ H.P.	570	.49
3.	Exhaust.	30-inch.	1.5 H.P.	1,612	1.53
4.	Supply.	45-inch.	3.5 H.P.	3,815	3.35
5.	Supply.	50-inch.	4 H.P.	4,625	4.23
6.	Supply.	35-inch.	2 H.P.	2,040	1.83
Total	...	.....	...	...	11.94

STEERING GEAR.

The steering gear consists of a steam steering engine located in compartment A-12 (berth deck, forward), under the chart house and steering wheel, connected by steel-wire rope passing along under the gun deck to the rudder arc and a reverse-screw hand steerer located at the rudder head on the main deck.

The steering engine is an inverted twin-cylinder direct-acting steam engine with a float valve, driving a worm on the crank shaft which engages a worm wheel on a vertical shaft,

which shaft carries a pinion engaging a spur wheel on a second vertical shaft. This latter shaft has fixed near its upper end a two-grooved pulley, to which one end of each tiller rope is fixed so as to unwind one rope while winding the other. Outboard are leading sheaves adjustable for varying the tension of the tiller ropes. The tiller ropes are led tangentially and fixed to the tiller quadrant by pelican hooks, which latter must be detached when the hand steerer aft is shipped. There is a clamp at the rudder head for use while making the change from steam to hand gear and vice versa. Before reconnecting the tiller ropes to the tiller arc it may be necessary to slacken the ropes at the guide sheaves aforestated. The links of the hand steerer are of the sort in common use in the service, where the reverse-screw shaft is worked by an engine or by handwheels, and are thrown in and out of gear by hand and secured by removable pins in the usual way.

THE CAPSTAN.

There is one capstan forward of pilot house on main deck, driven by a twin-cylinder, simple, vertical engine, 7 inches by 8 inches, located nearly underneath, on gun deck, through a worm and worm-wheel gearing and vertical shaft. Under the capstan drum is a wildcat fitted with a band brake for taking the chain cable, which wildcat can be operated either by hand, through the capstan or be thrown out of gear. The capstan above the wildcat can be operated by hand, with capstan bars, or by the engine, by the adjustment of clutch keys for the purpose. There are two large chain bits forward and four leading bits on the capstan bed plate.

The bow anchors stow in the hawse pipes and weigh, 3,000 pounds each.

ELECTRIC EQUIPMENT.

The electric generating installation consists of two multipolar continuous-current generators each driven by a compound engine, both manufactured by the General Electric Company. Engine marks: type, CC-24-400; form, H-4. K.W. 24; steam, 100, revolutions, 400. They are vertical direct-acting, and have

cylinders of 6½ and 10½ inches diameter by 7 inches stroke. Diameter of piston rods, 1 $\frac{3}{16}$ inches. The starboard engine bears shop No. 2,228. The port engine bears shop No. 2,229.

The dynamos are of type, M.P.; class, 8-24-400; form, D; volts, 125; amp., 192; K.W., 24; speed, 400; shunt current, 4.1. The starboard dynamo bears shop No. 87,344; the port dynamo bears shop No. 87,345.

All the wiring is in conformity with U. S. Navy standards, in wrought-steel enameled-pipe conduits on the two-wire feeder system.

There is one searchlight located on a platform over and abaft the bridge. Its projector is 18 inches diameter. Other lights distributed through the ship as follows:

Berth, gun and main decks forward are lighted by twenty-four 16-candlepower lamps.

Berth, gun and main decks aft are lighted by fifty-nine 16-candlepower lamps.

The engine room is lighted by thirty-two 16-candlepower lamps including six portables. (Additional lights are being installed.)

The fireroom is lighted by twelve 16-candlepower lamps including three portables.

There are forty-four battle lights of 16-candle power, and twelve of 32-candlepower.

MAIN ENGINES.

There are two *main engines* located in one compartment abreast one another. They are vertical, three-cylinder, triple-expansion, direct-acting, with air pumps, main-feed, sanitary and bilge pumps attached. The original plans contemplated inward-turning screws, which were changed to outward-turning by exchanging the positions of the engines, shafting and propellers. This new arrangement placed the starting gear on the outboard sides and far apart, necessitating two men for working engines to signals.

The cylinders are of cast iron, unjacketed, the H.P. cylinder forward, and rest on wrought-steel cylindrical columns

and cast-iron box columns. The former columns are outboard, extending from cylinders to bedplate; the latter are bolted between the housings on top of condensers and lugs on cylinders. Brackets on the inboard columns support the reversing shafts. The cylinders and steam chests, including the bottom heads, are cast iron, one piece—the H.P. forward—and are bolted together. Steam enters the H.P. chest between the valve heads, and exhausts at their ends, passing around the H.P. cylinder in passages through the casting to the ends of the intermediate valve chest, exhausting from the middle through a 7-inch copper pipe to the L.P. steam chest.

The main pistons are made of cast steel, class A, and, with their followers, are finished all over. The followers are wrought steel. They are fitted with two packing rings each, of cast iron, tongued where cut obliquely, and retained in a fixed relative position by the heads of the bolts securing the tongue pieces engaging slots in projections on the pistons. No restraining lugs or distance pieces are provided, nor are any springs provided for setting out the rings.

The engine bed plates are of cast iron, each in one piece, resting on three longitudinal seatings worked in the ship's structure on top of the ship floors, and two intercostal girders. No diagonal braces are provided for these engines, as the rigid construction of the condenser and inboard columns seems all sufficient for the small masses involved.

The main valves are one for each cylinder, of cast iron, and of form as follows: For the H.P. and I.P., single-ported piston valves with equal heads; for the L.P., flat double-ported slides. No balancing is provided except by a gravity-balance steam piston working off the low-pressure valve, the top connected permanently with the main condenser by a pipe. All main valves are operated directly by Stephenson double-bar links and eccentrics. There is no independent cut-off gear other than adjustable connections between the reach rods and reversing-shaft arms. The slots in the reversing-shaft arms are graduated by marks indicating the mean of the up-and-down points of cut-off in inches from beginning of stroke.

Main Valve Packing.—The intermediate piston valves only are fitted with packing rings, and suitable lugs are provided to prevent them from overriding the seats when the valves are disconnected. These packing rings are in one piece for each end of the piston valve with a single parting for expansion and adjustment. At the parting are lugs with bolts and jam nuts. All the main H.P. and I.P. valve chests are fitted with cast-iron liners. The cylinders have no liners.

The piston and valve rods are packed with United States metallic packing of latest design, consisting of a combination of beveled white metal rings and spiral springs. It was fairly tight during the trial.

The crank shafts are solid class A forgings, excepting an axial hole through each crank pin, and are each in one piece. The cranks are set at 120 degrees apart. Between the intermediate and L.P. cranks is a worm wheel for turning the shaft by hand gear. There are six crank-shaft bearings, one pair to each crank, with two bolts each. Each bearing is composed of a bottom brass, two distance pieces and a cap. The cap is of cast steel, class A, the other parts of composition. The cap and brass are lined with white metal and provided with oil channels, into which the oil enters near the side.

The piston rods are of high-grade steel forgings, solid, accurately ground and polished. The upper end of each is secured to the piston by a shoulder, taper, nut and split key. A dowel pin between piston and rod prevents turning between the parts. The lower end of the piston rod includes the crosshead, to which is bolted a composition slipper of manganese-bronze, white-metal faced both sides. The end of the piston rod or gudgeon carries a pair of brasses, white metaled, with a two-bolt cap for the connecting-rod pin. All crossheads, piston and connecting rods are of the same dimensions for the different sections, *i. e.*, cylinders.

The connecting rods are of high-grade steel forgings, forked at one end to span the crosshead, the crosshead pin shrunk in, secured from turning by screw keys and fitted with white-metaled crank-pin brasses at the other. The lower end of the

connecting rod is flat, against which are bolted the crank-pin brasses and steel binder by two high-grade steel bolts. The intermediate crosshead journals or pins differ from the others in the presence of wrist pins for the air-pump beam connecting links. For one-third of their projected area the crosshead pins are flattened on their (inboard and outboard) sides.

The eccentrics are of cast iron, each in two parts bolted together, rabbetted at each side for the flanges of the eccentric straps. Each backing eccentric is secured to the shaft by a key and each go-ahead eccentric likewise, with additional adjusting pieces between its key and the eccentric.

The eccentric straps are of composition, lined with white metal, which latter is dovetailed and hammered in place. The eccentric rods of forged steel have forked ends at their link connections and T-shaped heads where bolted to the eccentric straps. The link wrist-pin brasses are adjustable.

The reversing gear for each engine consists of a steam cylinder 5 by 9 inches secured to the intermediate cylinder casting, a reversing shaft resting in bracket bearings on inboard engine columns, an eccentric secured to the reversing shaft, connecting and reach rods, and manipulating lever with locomotive latch at working platform. The reversing-engine piston is of cast iron fitted with cast-iron snap rings. The reversing-engine valve is a plain D slide, which receives its primary motion from the reversing hand lever and its secondary motion from an eccentric on the reversing rock shaft. No hand-reversing gear is provided. The engines are fitted with a hand turning gear consisting of an adjustable steel worm on a short shaft with square end, engaging a worm wheel on the crank shaft between the intermediate and low-pressure cranks. A ratchet wrench working horizontally affords the necessary purchase. The worm is removable by means of a two-bolt steel binder.

The *main condensers* are of the surface-condensing type, steam outside the tubes, with shell of cast manganese-bronze of rectangular form in horizontal section side elevation and plan with cylindrical tops, flat bottoms and sides. The side of

each condenser nearest the crank shaft is inclined sufficiently to accommodate the crank circle. They are slightly longer than the engines, projecting aft beyond the engines only by the by-pass channel and valve on the after heads, and form the housings for the inboard sides of the main cylinders. On the tops and integral with the shells of each condenser are the following nozzles: Circulating-water outlet, exhaust steam, safety valve and two 6×4 handholds; also on top are bosses for reverse-gear brackets and housings for the air, feed, bilge and sanitary-pump rock-shaft pillow blocks. The thickness of the shell is seven-sixteenths of an inch, stiffened by rectangular ribs where necessary. A division plate at the after ends causes the circulating water to pass forward through the lower nest of tubes and aft through the tubes of the upper nest. For the purpose of pumping from the bilge by the main circulating pumps a by-pass valve and pipe of rectangular section connects both sides of the division plate. The auxiliary exhaust and the bleeder pipes are connected with the main exhaust pipe. There is a connection of four inches diameter for the air-pump suction and one two and one-half inches diameter for a pipe between the condensers at bottom. Joining this latter pipe is a permanent steam connection with the bleeder-pipe cross, one inch in diameter, for boiling out the condenser. The lower valve on this connection is under the floor plate amidships in line with both circulating pumps. A horizontal perforated brass plate forms a diaphragm within, completely across and five and three-quarter inches from the top of the condenser, leaving a clear space over the ends to scatter the exhaust steam. The perforations, $\frac{1}{2} \times \frac{3}{4}$ inch pitch, occur for forty-seven inches by seventeen and a quarter inches forward of the exhaust nozzle; under the exhaust nozzle the plate is not perforated. The tubes, in two nests are straight, supported by a middle diaphragm and packed in rolled tube plates of same composition as tubes by screwed glands over cotton tape. The condensers were tested under thirty pounds hydrostatic pressure, and the safety valve afterwards set at 25 pounds.

Particulars of condensers.

Number of condensers	2
Steam inside or outside tubes	Outside.
Cooling surface, square feet, starboard	678.716
port	678.716
total	1,357.432
Ratio of total cooling surface to total heating surface	1 : 3.0638
Number of tubes to each condenser	488
Diameter of tubes, outside, inch	$\frac{5}{8}$
Length of tubes, feet and inches	8- 6
Spaced, inch	$1\frac{1}{16}$
Composition	Cu. 70, Zn. 29, Sn. 1.
Thickness of tube plates, inch	$\frac{3}{4}$
Ferrules for packing, outside diameter	$\frac{3}{4}$-inch, 18 threads.

An air pump located inboard of each condenser is worked off the intermediate crosshead of each engine, as also are sanitary, bilge and main feed pumps, arranged as follows:

Forward of both air pumps and attached to arms on the main-pump rock shafts are the feed pumps; abaft the air pumps are the bilge and sanitary pumps—the former to the port engine, the latter to the starboard engine. The bilge, sanitary and feed pumps are of the differential plunger type with water-packing grooves. All pumps on main engine and the main and auxiliary feed, fire and bilge pumps have spring safety relief valves. The air pumps are of the vertical-bucket closed-top type with sheet-bronze valves. All these engine pumps are of composition of the Navy standard in their chambers, buckets, plungers and valve chests. Each pump has an independent crosshead attached by double-link connections of steel rods and composition bearings to the pump rock-shaft arms. The air-pump guides are bolted to and the other pump guides are cast with their respective top bonnets. The main-pump rock shaft is supported by the pillow blocks, of two parts each, straddling each pump arm, resting on housings cast on the top outboard side of each condenser.

The thrust, stern sleeve and *propeller shafts* are each in one section, constituting with the crank shafts the main shafting of the vessel. They are all (excepting the crank pins) of solid class-A forgings, finished all over to a diameter of 5 inches

before encasing. The propeller, stern-tube and thrust-shaft after couplings are double-taper steel sleeves, with two "feathers" or longitudinal keys, 90 degrees apart each. Two through keys at each sleeve coupling, one through each section end, bind the sections together. At these couplings the ends of the shafts are one-half inch apart. Screw keys between the through keys and sleeves secure the keys in place. Five-bolt flange couplings unite the crank and thrust-shaft sections.

The stern-tube and propeller shafts are encased by composition sleeves, pinned on with brass stop screws, spaced spirally. A fair water containing pitch encases the outboard shaft couplings. Each thrust shaft has five thrust collars.

There are two outboard brackets or struts and hangers to each shaft, of manganese-bronze, with sleeves lined by lignum-vitae strips, end of grain to the shaft. The stern and bracket sleeves are in halves longitudinally. The stern tubes have a bearing at each end.

The *thrust bearings* each consist of a pedestal and bearing proper. The bearing is fitted in a bored part of the pillow block, which latter is bolted to its seating, built in the ship's structure. It is in two parts, of cast iron, with white metal linings. The upper part is fitted to the lower with dowel pins and longitudinal recesses in the upper flanges of the pedestal, to which it is held down by steel bolts, body bound in the pedestal, and with slotted holes in the cap. The ends and side walls of the pedestal form an oil trough, from which lead oil holes to the collars and recesses. Stuffing boxes and glands are provided to retain the oil, and four adjusting screws in lugs on the pedestal provide longitudinal adjustment of the bearing. Forward of each thrust bearing on the same pedestal is a spring bearing, resting on and adjustable vertically by a broad wedge liner. This wedge is between the spring-pillow block and base of thrust-bearing block, adjustable by set screws on each side. A mandrel is provided for rebabbitting the thrust bearing.

Main Circulating Pumps.—Two compound engines drive two centrifugal pumps—one each side of engine room. The engines are very light, both cylinders and valve chests in one

casting, with forged finished-steel column supports, cast-iron bed plate, valve rock shaft, cranks at right angles, complete adjustability of all parts and fitted with indicator gear and counters. The H.P. piston is of cast iron, fitted with two snap rings, without follower, of cast iron. It is screwed on the piston rod, the end of which is riveted over in a countersunk recess in piston. The L.P. piston is of cast iron fitted with two rings and a follower. The piston and follower are secured to the rod in the manner described for the H.P. piston with the addition of rivets through piston and follower instead of follower bolts. The H.P. valve is of the solid-piston type. The L.P. valve is a D slide with a composition spring-borne tongue fitting into its back, as a guide and to press the valve against its seat.

The pumps are two, one to each engine, of composition throughout. Each runner has eight blades, curved at the tips. The runner is keyed by a tobin-bronze key to a tobin-bronze shaft. A four-bolt flange coupling joins engine and pump shafts. The casing is in two parts joined together by a nearly horizontal flange joint and brass bolts. The lower half has, at and near its base, three nozzles, one pair connecting with the sea injection and with the main drain with interlocking valves; the other nozzle connecting with the circulating water pipe to the condenser. Suitable channels in lower section of casing lead the suction to both sides of the runner at center. There is a lignum-vitae bearing, covered by a composition circular plate, for the after end of the runner shaft. A stuffing box and gland only is provided for the forward end, as the engine shaft bearings are adjacent.

CIRCULATING PUMP DATA.

Diameter of H.P. cylinder, inches	3
L.P. cylinder, inches	6½
Stroke, inches	5
Diameter of piston rods, inch	⅝
pump runners, inches	18
Width of pump runners, inches, at periphery	2¾
hub	4¾
Diameter of inlet nozzle, inches	5½
outlet nozzle, inches	5½

There are two *feed and filter tanks* of ⅛-inch galvanized sheet steel in the engine room, one forward of each engine. The capacity of each is 245 gallons, about, including filter boxes. They are clothed with hair felt and sheathed by planished iron. The filter compartments, four in number, occupying about one-fourth the total of the tanks, are worked in the upper part of the tank with partitions watertight at the sides and bottom, such as to cause the water from the air pumps to flow through filter material from top to bottom of each filter compartment, passing between them from the bottom of each to the top of the next. Each tank is provided with a safety relief valve on the first filter compartment and a vapor pipe from both the first filter compartment and side near top of feed tank. The top and removable cover are of composition. At the inboard end of the top of each tank are four nozzles or openings for vapor pipe to deck, spring relief valve, connection from cylinder relief valves and for air-pump discharge. The vapor pipe and cylinder relief-valve pipe openings are 2½ inches diameter; the air-pump discharge and the cylinder relief valve to tank are 3-inch diameter each. The spring relief valve from tank discharges into a 3½-inch overflow pipe with open funnel top to bilge. At the bottom of each tank is a ¾-inch drain cock. A 3-inch suction opening to feed pumps at bottom inboard side of each tank is controlled by an automatic ball float valve, or may be kept open by hand lever and latches to valve mechanism. All filter compartments can be drained into lower reservoir by suitable ½-inch valves and pipe on the front of tanks. The tanks are connected by a pipe and valve between the two tanks for the purpose of equalizing the water level in both or for using either tank separately. As the main and auxiliary cylinders are not equipped with oil cups, no considerable cylinder oil is likely to be found in the filter tank other than may get into the L.P. cylinders by way of the packing around the piston rods. There are no independent filters on the feed-water system.

The main feed pumps, one worked off each main engine, are of composition of the differential plunger type and have the following dimensions:

Diameter of plunger, greatest, inches	$3\frac{5}{8}$
least, inches	$2\frac{9}{16}$
Stroke, inches	4.95

The *air pumps* are two, of the bucket plunger type, one worked off each main engine, and have the following dimensions:

Diameter, inches	12
Stroke, inches	6.9
Diameter of pump rod, inches	$1\frac{1}{4}$

The sanitary and bilge pumps attached are of the differential plunger type and have the following dimensions:

Diameter of plunger, greatest, inches	$5\frac{1}{2}$
least, inches	$3\frac{3}{8}$
Stroke, inches	4.95

MAIN ENGINES.—PARTICULARS OF ONE ENGINE.

Cylinders, number for each engine	3
H.P., diameter, inches	9
I.P., diameter, inches	$15\frac{1}{2}$
L.P., diameter, inches	$25\frac{1}{2}$
Stroke of all pistons, inches	21
H.P. valves (piston), diameter, top, inches	$4\frac{3}{8}$
bottom, inches	$4\frac{3}{8}$
I.P. valves (piston), diameter, top, inches	8
bottom, inches	8
L.P. valves (slide), length, inches	21
width, feet and inches	2- $2\frac{1}{2}$
Valve stems, diameter at stuffing box, inches	$1\frac{3}{4}$
below stuffing box, inches	$1\frac{7}{8}$
through valve, inches	$1\frac{1}{2}$
L.P. valve, stem, diameter at balance pistons, inches	$1\frac{1}{4}$
Main steam pipe, diameter at throttle, inches	3
area of cross section, square inches	7.0686
Volume swept by H.P. piston, mean, per stroke, cubic feet	0.7731
I.P. piston, mean, per stroke, cubic feet	2.2197
L.P. piston, mean, per stroke, cubic feet	6.2065
Clearance of H.P. cylinder, per cent., top	19.83
bottom	19.64
I.P. cylinder, per cent., top	17.44
bottom	13.98
L.P. cylinder, per cent., top	9.75
bottom	6.64
Net area of I.P. to H.P. piston	2.871 : 1
L.P. to H.P. piston	8.0271 : 1

Piston rods, diameter, inches	2¼
length, feet and inches	3– 6⅝
Cylinder walls, H.P., I.P. and L.P., thickness, inch	⅞
Valve-chest liners, thickness, inch	½
Cylinder relief valves (one each end cylinder), diameter, inches	1½
Connecting rod, length, center to center, inches	52½
diameter, upper end, inches	2½
lower end, inches	2⅝
crosshead bolts (4), diameter, inch	⅞
Connecting-rod crank-pin bolts (2), diameter, inches	1½
Crosshead, surface, ahead, square inches	70
backing, square inches	35
pin, diameter, inches	3⅝
length of pin, inches	5½
Crank shaft, number of sections	1
diameter, inches	5
coupling discs, inches	10½
coupling bolts, number in one flange	5
diameter, inches	1¼
journals, diameter, inches	5
length, inches	5 and 7
length of sections, feet and inches	one piece.
maximum distance, beyond journals, feet and inches	1– 3
Crank pin, diameter, inches	5¼
length, inches	8
projected area, square inches	42
diameter of axial hole, inches	2½
webs, width, inches	6
thickness, inches	3⅝
Thrust shaft, diameter, inches	4¾ and 4⅝
length, feet and inches	19– 3¾
steady bearings, inches	11
collars, number on each shaft	5
diameter, inches	10
thickness, inches	1½
space between, inches	1⅝
total surface, both engines, square inches	598.75
Line shaft, diameter, inches	5
sleeve coupling, inches	8
length, feet and inches	16–10½
Propeller shaft, diameter over liners in strut bearings, inches	6
after stern-tube bearings, inches	6
in forward stern-tube bearings over liners, inches	6
length, feet and inches	25– 8 5/16
Shaft-sleeve couplings, diameter, inches	8
length, inches	20½
Stern-bracket bearings, diameter, inches	6

Stern-bracket bearings, length, feet and inches	1–11½
Stern-tube bearings, diameter each, inches	6
length, each, inches	30
bracket bearings, forward, diameter, each, inches	6
length, each, inches	17½
after, diameter, each, inches	6
length, each, inches	23½
Reversing engine, diameter of steam cylinder, inches	5
stroke, inches	9

VALVE DIAGRAM DATA.

Type of engines	Direct-acting triple-expansion.
Diameter of cylinders, inches	9, 15½, 25½
Stroke of cylinders, inches	21, 21, 21
Valve gear	Stephenson type with double-bar links.
Revolutions per minute	225
Piston speed per minute, feet	787.5

	H.P.		I.P.		L.P.	
Eccentricity, inches	1⅜		1⅜		1⅜	
Travel of valve, inches	3½		3½		3½	
Type, number and diameter of piston valves	One 4½ inches		One 8 inches		*	
Side of valve on which steam is taken	Inside.		Outside.		Outside.	
	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
Width of port, inches	1½	1½	1½	1½	2 × 1½	2 × 1½
Length of slide valves, inches	...	...	...	...	24¾	24¾
Steam lap, inches	1 1/16	1 1/16	1 1/16	1 1/16	1	1 1/16
Exhaust lap, inches	−3/32	3/64	−1/8	1/8	9/64	1/64
Angular advance, degrees	35	35	37	37	39	39
Steam lead (angular), degrees and minutes	7, 57	12, 25	8, 15	12, 45	9, 0	13, 5
(linear), inches	1/32	1/16	1/32	1/16	1/32	1/16
Cut-off, in inches	16.212	15.351	15.666	14.721	15.162	14.070
in decimal of stroke	.772	.731	.746	.701	.722	.670
Release, in inches	2.062	1.937	2 156	2 015	1.5	1.718
in decimal of stroke	.097	.093	.102	.096	.072	.081
Compression, in inches	1 66	1.80	2.015	2.156	3.406	2 86
decimal of stroke	.079	.084	.096	.102	.162	136
Steam opening, linear, from diagram, inches	1 1/16	1 3/16	1 1/16	1 3/16	2 × 1/2	2 × 1/2
area of, in square inches	10.74	12.17	18.12	20.65	43.31	48.70
Exhaust opening, linear, from diagram, inches	1¼	1¼	1¼	1¼	2 × 1½	2 × 1½
area of, in square inches	16.84	16.84	29.93	29.93	61 87	61.87
Steam velocity through ports, in ft. per minute	4,664	4,116	7,938	6,964	9,285	8,279
Exhaust velocity through ports, in ft. per min.	2,980	2,980	4,806	4,806	6,500	6,500
Velocity through main steam pipe in feet per minute						7,095
Velocity of exhaust to condenser in feet per minute						7,088

* One double-ported slide valve.

Data refers to valves in full gear "ahead" and full gear "backing."

A *feed-water heater* is located on the port side forward in engine room. It is composed of a composition cylindrical shell with flanges, to which are bolted at each end a tube-plate and a header. Expanded into the tube plates are one hundred thin, seamless-drawn ⅝-inch brass tubes, No. 14 B.W.G. thickness, 4 feet 3½ inches long. The heater is ar-

ranged with feed-pipe connections, so that the discharge from the main-engine pumps and independent main-feed pump may be passed through or by-pass the heater. The auxiliary exhaust-steam pipe is connected with the heater shell near its upper end through a 4-inch nozzle, the outlet for the exhaust steam being at the side near the bottom. Three baffle plates through which the tubes pass, of rolled naval brass resting on four lugs each, guide the steam across and among the tubes. Suitable air cocks and drain pipe are fitted. The headers and tube sheets are stayed together by nineteen $\frac{5}{8}$-inch tobin-bronze socket bolts with brass pipe sleeves at each end. The feed water enters the bottom header, passes through the tubes and out at the top header. The headers are of composition with feed-pipe nozzles cast on. An adjustable automatic-spring relief valve on the auxiliary exhaust pipe at each main condenser regulates the exhaust pressure. *There are no auxiliary condensers.*

The *propellers* are of the three-bladed bent-back type, of manganese-bronze, true-screw, polished, tinned and repolished. The blades are cast with the hub. Two keys and nut on the taper end of the propeller shaft secure each propeller to its shaft. Over the nut is a composition cap or fairwater with watertight joint filled with tallow. These propellers were measured and fitted to guides or templets of thin wood, carefully arranged to conform to the drawings, and much time and care was devoted to making them true in every respect, including balancing. The following are the principal dimensions, etc., for one propeller.

PROPELLER DATA.

Number of blades	3
Diameter of propeller, feet and inches	7- 3
Pitch of propeller, feet and inches	6-10
Expanded area, square feet	15
Projected area, square feet	12.642
Ratio of pitch to diameter	1 : 1.06
Ratio of total expanded blade area to immersed midship section at trial draught	1 : 11.6
Ratio of expanded blade area to disc area	1 : 2.752
Angle of inclination of generatrix with axis of hub, degrees	77½

The boilers are two in number, separated by an athwartship fireroom eleven feet wide, in one compartment. They are the Babcock & Wilcox Co.'s *Alert* type, and were constructed by that company at their works at Bayonne, N. J. As a complete description of this type of boiler has recently been published in the JOURNAL (Vol. XVI, p. 1223), a few particulars will suffice here, as follows:

FOR ONE BOILER.

Diameter of drum, inside, inches	42
Length of drum, neglecting heads, feet and inches	9- 6⅝
Thickness of drum plate, inch	13/16
Number of elements	15
Number of tubes, ordinary (2-inch)	108
upper, circulating (4-inch)	15
lower, circulating (4-inch)	2
side boxes, each side, rectangular	4
Length of tubes between headers, feet	9
over all, feet and inches	9- 2
Tube-heating surface, square feet	1,970.7
Plate-heating surface, side boxes and drum, square feet	108.8
Aggregate heating surface, square feet	2,079.5
Height of smoke pipe above grate, feet	60.33
Area through first section for gases of combustion, square feet	10.36
second section for gases of combustion, square feet	10.2
third section for gases of combustion, square feet	8.6
uptake section for gases of combustion, square feet	8.2
Sectional area of smoke pipe (37 inches diameter), square feet	7.4666
Ratio of heating to grate surface, fireside considered	41.68 : 1
Number of tubes, 2-inch, in intermediate sections	28
right-side section	23
left-side section	21
lower sections	2
4-inch tubes in each lower section	1
Thickness of header tube sheets, inch	½
Number of plates in shell of drum	1
Thickness of butt straps and tube-hole reinforcing, inch	13/16
Depth of header, outside, inches	5⅝
Width of header, outside, inches	6½
Weight of one boiler, complete, pounds	30,045

Boiler mountings include for each boiler a duplex 3-inch pop safety valve, two water gauges (one reflex and the other a tube protected by plate-glass shields), both in Dewrance fittings, with safety checks and quick-closing cocks by chains

from fireroom; one main-feed stop and check valve, one auxiliary-feed stop and check valve; one main-steam stop valve with automatic steam closing appliance; one surface-blow valve and two bottom-blow valves and three gauge cocks. Each steam drum has near its highest part and at middle a fitting for two steam gauges, air valves and testing connection.

At the rear of each grate is a sprinkler for extinguishing fires in furnace, operated by a pipe conection and valve with the fire main at the back end of each boiler.

The ash pans are galvanized-steel and are shallow. The ash pits are fitted with automatically-closing doors which can be quickly lifted out and laid away if desired.

Forced Draft—Closed Fireroom System.—By closing trap doors over fireroom hatch on main deck, bulkhead doors between drum room and gun deck, and one between engine and firerooms, and starting the forced-draft fans over fireroom on gun deck level, the forced-draft system may quickly be put in operation and as quickly be discontinued. As the bunkers ventilate through permanently open ducts into the casing around smoke pipes, either the bunker doors should be kept closed, the exhaust nozzles in bunkers be covered, or a slight loss of efficiency will result. No gates are provided for the bunker-exhaust ducts. Two 40-inch centrifugal fan blowers, driven by 3 × 3-inch simple twin-cylinder engines with cranks opposite, are located on a level with the gun deck, over the fireroom, one forward on port side, the other aft on starboard side. They take air from two fireroom ventilators in their vicinity, which serve also as ash hoists.

The blower engines for forced draft are inverted direct-acting, mounted on forged-steel columns, have piston valves, are equipped with sight-feed and centrifugal oiling gear, and have long bearings. They perform excellently well on all occasions without heating of bearings at any speed. An indicator card from one of these engines was taken at 700 revolutions per minute, and at times their speed exceeded a thousand revolutions per minute without apparent injury or excessive wear.

Number of blowers	2
Type of blower engine	Vertical, inverted, twin.
Diameter of cylinder, inches	3
Stroke, inches	3
Diameter of piston rod, inch	$\frac{5}{8}$
fan, inches	40
Width of fan at tips, inches	5
Area of induction nozzle, square feet	3.72 to 2.18
eduction nozzle (periphery of fan × width), square feet	4.35

Ash Hoists.—The fireroom ventilators are arranged with suitable guides and doors, as customary.

An engine of same form and size as the blower engine, *i. e.*, 3 × 3, is secured to the main-deck fireroom-hatch framing, amidships, and a spur gear on the engine shaft engages one larger on a drum shaft extending athwartships, abaft the engine and beyond both ventilators. A drum at each end of this shaft, mainly without, but whose periphery is within each ventilator, winds and unwinds the ash whips. A clutch on the drum shaft permits either drum to be used independently. A small hand lever starts the hoister, which latter stops automatically when the bucket reaches either end of the hoist. It was found desirable to introduce a reducing valve in the steam pipe to and near the ash-hoist engine, which provided safe working at the extremes of sea and port pressures, as the engine worked too violently under full-steaming pressures.

A bucket of ashes can be hoisted from fireroom to main deck in two seconds.

Trolley rails overhead on main deck, fitted with slides and hooks and unhooking levers, lead to ash chutes on each side of ship.

EVAPORATING AND DISTILLING PLANT.

Two evaporators of contractors' design and two distillers of Bureau of Steam Engineering design are installed; the former in the engine room, one each side, aft; the latter in the engine room hatchway, forward side. Each evaporator consists of a steel cylinder, with longitudinal axis vertical. Horizontal straight brass tubes expanded into headers are fitted into a composition rectangular frame in lower part of shell. Two

steel diaphragms or baffles are in the steam space; the upper diaphragm, twelve inches from the top, is a perforated plate, ¼-inch holes; the lower is a conical plate, with a large central opening, hooded. The baffles are twelve inches apart. The inverted hood under the lower baffle is three inches below it, and dished on its edges downward. The upper head is inserted, and is dished to a radius of about five feet. The bottom is flat, inserted, stiffened by two angles, and riveted to shell with flange external. Each header has two vertical stiffening ribs and stay bolts.

DATA FOR EACH EVAPORATOR.

Diameter shell, inside, inches	33
Length over all, feet and inches	5– 9
Thickness of shell, inch	$\frac{5}{16}$
heads, inch	$\frac{3}{8}$
Number of tubes	72
Diameter of tubes, inches (outside diameter)	2
Thickness of tubes, No. B. W. G	12
Length of tubes, inches	$28\frac{5}{8}$
between tube plates, inches	$26\frac{3}{4}$
Pitch of tubes, inches	$2\frac{1}{4}$
Thickness of tube plates, inch	$\frac{7}{8}$
header plates, inch	1
Number of stay bolts in each header	16
Diameter of stay bolts in each header, inch	$\frac{7}{8}$
Material of headers	Composition
stay bolts	Tobin bronze.
Heating surface, square feet	84.04
Capacity of evaporators combined, nominal, gallons	3,500
Actual output on test, rate per diem, gallons	4,034

Distillers.—Two of Bureau design, each with 32 square feet cooling surface, are located side by side on forward bulkhead of engine-room hatch between main and gun-deck gratings. As this type of distiller is so well known in the service, a slight description only should be essayed: Each consists of a vertical cast-iron cylinder, with flanges for bolting tube plate and header together at top end and a stuffing box and packing gland and bolts at the bottom end. Straight tubes are expanded at one end into a composition tube sheet, which is secured between the upper flange of the cylinder shell and the aforesaid header. The lower ends of the tubes are expanded

into a cylindrical box or header, which latter is free to move by the expansion of the tubes in the stuffing box.

The vapor from the evaporators is admitted through a nozzle close to the top at side of shell, around (*i. e.*, outside) the tubes, while the circulating water is introduced at the bottom header, and, passing through the tubes, makes its exit, at the top of the top header.

At the vapor nozzle within the distiller shell is a perforated galvanized sheet-iron baffle ring surrounding the tubes at the upper end, with an axial length of $3\frac{1}{8}$ inches and inside diameter of $7\frac{3}{4}$ inches. This baffle ring is flanged at both ends to $8\frac{15}{16}$ inches diameter, which is nearly the inside diameter of the distiller shell. By the upper flange the baffle ring is secured by four small brass screws to the upper tube plate. The perforations in the baffle plate are 64 of $\frac{1}{4}$-inch diameter.

The distillate is drawn off at a nozzle at the lower side of shell, and flows by gravity or may be pumped to the ship's tanks or through the fresh-water system. Intermediate between the distillers and fresh-water pump adjacent is a small galvanized-iron reservoir of about three gallons capacity, with removable cover, which can be filled and emptied through the fresh-water pump suction from distillers. A water meter is on the same line of pipe between distillers and said reservoir.

DATA FOR EACH DISTILLER.

Diameter, shell, outside, inch	$9\frac{3}{4}$
inside, inches	9
Length, shell, between flanges, outside, feet and inches	4– $3\frac{3}{8}$
distiller over headers, feet and inches	4–$10\frac{5}{8}$
Thickness, upper tube sheet, inch	$\frac{3}{4}$
lower tube sheet, or header sheet, inch	$\frac{5}{8}$
Number of tubes	46
Diameter of tubes, outside, inch	$\frac{5}{8}$
Length of tubes (over all), feet and inches	4– $1\frac{5}{8}$
between headers, feet	4
Cooling surface, outside, square feet	nearly 30.2

OILING ARRANGEMENTS.

Galvanized steel oil tanks for storage of lubricating, cylinder and lamp oil are located in the after end of the engine room, secured to athwartship bulkhead, one for each kind of oil, the form and capacity of which are, for the lubricating

oil: rectangular, 66×18×60 inches inside measurements, capacity 300 gallons; for the cylinder and lamp oils, one each rectangular, 18×18×36 inches, of 50 gallons capacity each.

The above-named tanks are filled through pipe from gun deck. A gravity system of filling the oil cups on the engine is supplied from a reservoir, cylindrical tank, of about 10 gallons capacity, secured between the distillers and feed-water pump in the engine-room hatch. A hand pump near by with piping from main oil tanks is fitted for filling this distributing tank.

Centrifugal oilers are attached to each crank pin of main and auxiliary machinery. Sight-feed oil boxes of polished cast brass are fitted each with twelve adjustable oil-feed valves and tubes for wicks, so that either or both wicks and valves may be used.

Each box contains swash plates, and each drip tube leads through a vertical glass tube, suitably dust proof, to the copper pipes to the different bearings.

Each main engine is fitted with four of these sight-feed distributing boxes. These boxes are supplied from a ½-inch brass pipe leading from the gravity tank fore and aft over the engine. This oil-distributing main is connected by a vertical pipe with a horizontal ⅜-inch brass pipe attached to the outboard side of condenser, supplying the oil cups on main crank-shaft bearings, of which there are six, to each engine.

The eccentric straps and similar oscillating gear are oiled through long, rectangular, open-top cups filled with curled hair, the supply of oil coming from the aforesaid distributing boxes about the top of the cylinders.

COOLING-WATER SERVICE.

A complete cooling-water service for all engine bearings, including line, shaft and thrust bearings, is installed in the usual manner. The cooling-water main supply is taken from the condenser-circulating outlet chamber, from the intake to circulating end of condenser, or may be forced by a hose connection from the fire pump in engine room. Spraying nozzles

PUMPS AND CONNECTIONS.

No.	Pumps	Size (inches).	Kind.	Suction pipes from—		Discharge pipes to—		Location.
				Inches.		*Inches.*		
2	Main air	12 dia. ; 6.9 stroke	Worked from I.P. crosshead	4	Condenser	3	Main feed tanks	1 on each engine.
2	Main circulating	Centrifugal	Vertical compound, 18-in. runner	5½	Sea	5½	Cond's'r and overboard.	1 for each condenser.
				5½	Main drain.			
				5½	Bilge.			Each side E. R.
2	Hotwell	4½ x 6 x 6, 95 gals	Vertical piston, double-acting, single, light service.	2½	Either air-pump suction	2½	Main feed tanks	In engine room.
				2½	C. & R. pipes from ship's sides.	2½	Reserve feed-water tanks.	Forward bulkhead between feed tanks.
				2½	Reserve feed-water tank.			
2	Main feed (attached)		Worked from I.P. crosshead	2	Feed-tank connecting pipe.	2	Main feed discharge main.	1 on each engine.
1	Main feed (independent).	9 x 5½ x 8, 90 gals	Piston plunger, double-acting, single	2½	Feed-tank connecting pipe.	2½	Main feed discharge main.	In engine room, port side.
1	Auxiliary feed (independent).	9 x 5½ x 8, 90 gals	Piston plunger, double-acting, single.	2½	Feed-tank connecting pipe.	2½	Auxiliary feed main.	In boiler room.
				2½	Auxiliary drain	2½	Fire main.	
				2½	Fire room bilge	2½	Overboard.	
				2½	Sea.			
				2½	Bottom blows	2½	Hose connection.	
				2½	C. & R. suction.			
				1½	Hose connection.			
1	Fire and bilge	8 x 6 x 12, 147 gals	Vertical piston, double-acting, single.	3	Sea	3	Fire main	Engine room.
				3	Main drain	3	Overboard.	
				3	Auxiliary drain	2½	Hose connection.	
				3	Drainage manifolds.			
				3	Engine-room bilge.			
				3	Crank pits.			
				1½	Hose connection.			
1	Bilge	3½ and 5½ diam. ; 4.95 stroke.	Worked from I.P. crosshead	3	Engine-room bilge	3	Overboard	On starboard engine.
				3	Crank pits, each engine.			
1	Sanitary	3½ and 5½ diam. ; 4.95 stroke.	Worked from I.P. crosshead	3	Sea	3	Flushing system	On port engine,
1	Distiller circulating	6 x 8 x 12, 261 gals	Vertical piston, double-acting, single, light service.	4	Sea	4	Distiller	Where directed.
						2½	Flushing system	
1	Evaporator feed	3½ by 2½ by 4	Light service		Sea		Evaporators	After bulkhead.
					Distiller, circulating water discharge.			
1	Distiller fresh water	Of approved capacity ; 3 x 2½ x 4.	Vertical piston, light service		Distillers, gravity discharge.		Fresh water tanks. Main-feed tanks. Distiller gravity discharge.	E. R. hatch.
1	Brine	3 x 2½ x 4	Light service		Bottom of evaporators		Overboard	After bulkhead.

are fitted over each crank-shaft bearing, crossheads and eccentrics.

ENGINE TRAMS.

To locate the proper (original) position of the main valve when the engine under consideration is "on the center," trams are made, two for both engines; also a tram for placing any crank "on the center," and bridge trams for locating the true (original) position of the crank-shaft journals in reference to their pillow blocks were fitted and neatly stowed in boxes for safe keeping. For the *Dubuque* and *Paducah* these latter trams proved to be exactly alike in all dimensions, so that one might be used for either ship, indicating a close or similar adherence to the drawings in both cases, in this particular at least. Generally speaking, the workmanship on these engines is excellent.

THE TRIAL OF U. S. S. *DUBUQUE.*

On the 5th day of May, 1905, the *Dubuque* was steamed from the contractors' works to the Navy Yard, New York, where the next day the ship was coaled with government coal for the purpose of bringing the ship down to the normal draught. Contractors' coal for the trial had been previously placed on board, weighed in bags, for convenience in tallying during the trial. This latter coal was Pocahontas hand-picked, and of excellent quality. It was stowed on the gun deck in temporary bins constructed for the purpose, and during the trial was taken into the fireroom through the bunkers, in bags, as wanted. On the 7th of May the vessel steamed to Newport, R. I., through the Sound, making the distance in eleven hours against a tide nearly all the way. The machinery on this run performed perfectly. On this run indicator cards were taken from the auxiliary machinery.

After taking aboard water in both trimming tanks, reserve water tanks and in water bottoms, to bring the vessel to a mean draught of 12 feet 3 inches, the progressive speed runs were made over a measured mile in Narragansett Bay on the afternoon of the 10th. A record of these runs is shown in the table and by the speed curves which follow.

Table No. 1.—U. S. S. *DUBUQUE*.

Data from progressive runs over measured-mile course off Newport, R. I., May 10, 1905.

Number and direction of run.	Time on course.	Revolutions of main engines.					Speed in knots per hour.	Indicated horsepower.										
		To make 1 knot.		Per minute.				Starboard engine.				Port engine.				Total both main engines.	Auxiliaries in operation (estimated).	Total all machinery.
		Starboard.	Port.	Starboard.	Port.	Mean.		H.P. cylinder.	I.P. cylinder.	L.P. cylinder.	Total.	H.P. cylinder.	I.P. cylinder.	L.P. cylinder.	Total			
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
1–S.	6.6233	794.9	882.4	120.1	133.2	126.6	9.06	24.34	29.91	17.82	72.07	28.08	40.82	29 47	98.37	170.44	26.67	197.11
2–N.	8.5233	1,176.2	1,124.6	138	131.9	135	7.04	37.89	46.60	23.83	108.34	27.59	32.84	17.79	78 22	186.56	25.38	211.94
3–S.	5 7333	883.5	894.5	154.1	156	155.1	10.464	54.39	6x.51	33 26	148.16	53.02	53.79	39.56	146.37	294 53	25.47	320
4–N.	6.6733	1,095.2	1,113.1	164.1	166 8	165.5	8.991	64.36	76.63	52.24	193.23	66.24	71.89	54	192.13	385.36	27 08	412.44
5–S.	5.3167	963.5	935.4	181.2	175.9	178.6	11 284	88.24	109.4	83.1	280 74	73.01	78 34	61.69	213.04	493.78	29.5	523.28
6–N.	5.49	1,069.4	1,111	194.8	202.4	198 6	10.93	91.68	128 7	98.8	319.18	110 14	121 16	109 2	340.5	659.68	34.86	694.54
7–S.	4.74	1,000.1	1,c24 5	211	216.2	213.6	12.658	127.6	169.8	148	445.4	135.67	163 58	154.57	453.82	899.22	38.8	938 02
8–N.	4.8367	1,091.7	1,102	225.7	227.8	226 8	12.405	143.8	207 5	199.7	551	158.58	181.08	180.68	520.34	1,071.34	45 8	1,117.14
9–S.	4.4717	1,055.9	1,079	236.1	241.3	238 7	13.417	181.3	248.7	262.4	692 4	203.08	231.11	252.58	686.77	1,379 17	75.06	1,454.23
10–N.	4.53	1,083 4	1,107.4	239.4	244.5	241 8	13.245	164.3	256.8	271.3	692 4	197.38	240.02	260.54	697.94	1,390.34	56.11	1,446.45
11–S.	4.5133	1,068.2	1,091.8	236.7	242	239.3	13.294	154 7	259.6	258.6	672.9	198.92	229.46	246.78	675.16	1,348.06	58.86	1,406.92

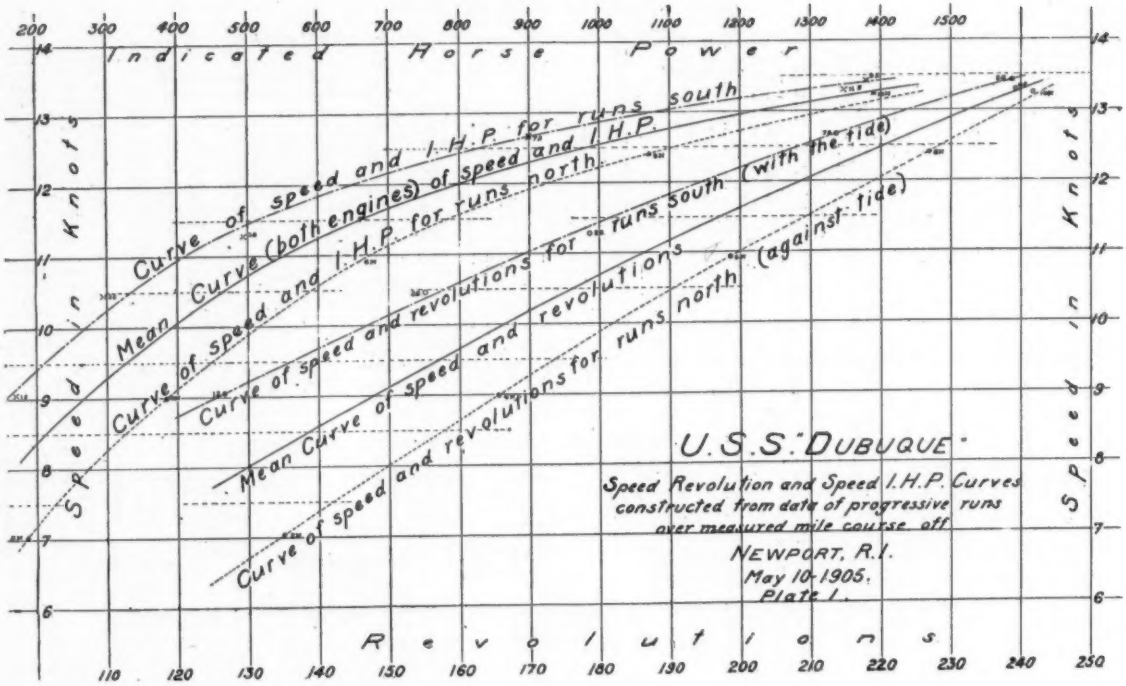
Table 1-A.—TRIALS U. S. S. *DUBUQUE*.—HORSEPOWER OF AUXILIARIES.

No. of runs.	Circulating pump.	Main feed.	Auxiliary feed.	Bilge.	Water service.	Blowers (2).	Dynamos.	Steering engine.
1	S. 0.39, P. 0.48	5	1.5	.....	0.5 (estimated).	S. 1.4, P. 1.4	15	1
2	S. 0.39, P. 0.49	4.2	1	.....	0.5 (estimated).	S. 1.4, P. 1 4	15	1
3	S. 0.48, P. 0.49	4.4	1.2	.....	0.5 (estimated).	S. 1.3, P. 1.1	15	1
4	S. 0.5, P. 0.48	4.8	1.4	1 (estimated).	0.5 (estimated).	S. 1.3, P. 1.1	15	1
5	S. 0.5, P. 0.4	4.8	1.3	1 (estimated).	0.5 (estimated).	S. 3.2, P. 2.5	15	1
6	S. 0.37, P. 0.39	5	1.3	1 (estimated).	0 5 (estimated).	S. 5.1, P. 5.2	15	1
7	S. 0.7, P. 0.4	7.2	1.3	1 (estimated).	0.5 (estimated).	S. 5.7, P. 6	15	1
8	S. 0.73, P. 0.77	13.3	1 4	1 (estimated).	0.5 (estimated).	S. 6.1, P. 6	15	1
9	S. 0.78, P. 0.78	17	1.6	1 (estimated).	0.5 (estimated).	S. 11.5, P. 11.5	15	1
10	S. 1.02, P. 1.09	13.1	2	1 (estimated).	0.5 (estimated).	S. 10.7, P. 10 7	15	1
11	S. 1.03, P. 0.83	16	2.1	1 (estimated).	0 5 (estimated).	S. 10 7, P. 10.7	15	1

Table No. II.—U. S. S. *DUBUQUE*.

Data from Progressive Runs over Measured-Mile Course off Newport, R. I., May 10, 1905.

Number of run.	Time at beginning of run P. M. (approximate).	Which engine.	Steam pressures.					Mean effective pressures.				Vacuum, inches.	Cut-off in cylinders from beginning.			Temperatures.						Revolutions or double strokes.						
			In boilers above atmosphere.	At engine above atmosphere.	H.P. steam chest, absolute.	In first receiver, absolute.	In second receiver, absolute.	H.P. cylinder.	I.P. cylinder.	L.P. cylinder.	Referred to low pressure.		H.P.	I.P.	L.P.	Outside air.	Engine-room working platform.	Fireroom working level.	Injection.	Discharge.	Hotwell tank.	Feed.	Circulating pump.	Main-feed pump.	Auxiliary-feed pumps.	Air-pump discharge temperature.	Water-service pump.	Bilge pump.	Fire-room blowers.	Air pressure, inches water.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
	h m																													
1	2–22	St'b'd	170	178	85	20	8	31	13	2.75	11.13	25	H.P., 0.751.	I.P., 0.723.	L.P., 0.695	Outside air, 75.	85	108	50	66	83	162	210							
		Port	178	155	80	24	14	32.25	16	4.1	13.68	28					82		45	76	75		220	30	9	76	18		350	
2	2–45	St'b'd	165	178	89	27	9	42	17.63	3.2	14.55	25.5					87	119	50	68	93	180	210			75			350	
		Port	165	180	75	20	9	32	13	2.5	10.99	28					76		45	69	97		225	24	5	79	17		350	
3	3–12	St'b'd	168	168	120	37	10	54	20.5	4	17.82	25.7					87	119	50	70	89	207	224	26	6	70	17		325	
		Port	165	173	105	30	9	52	18	4.7	17.39	28.5					76		48	76	88		225			90			300	
4	3–30	St'b'd	178	168	135	43	13.5	60	24.38	5.9	21.83	26.2					87	117	49	74	84	190	220	28	8	70	12	50	325	
		Port	175	174	125	35	11	60.75	22.5	6	21.35	28.5					76		50	81	83		220			84			300	
5	3–47	St'b'd	165	163	165	61	23	74.5	31.5	8.5	28.71	26.5					87	118	49	83	88	179	230	28	7	72	12	50	600	.2
		Port	160	170	134	40	13	63.5	23.25	6.5	22.45	28					77		50	80	84		215			84			550	
6	4– 4	St'b'd	180	188	185	70	25	72	34.5	9.4	30.37	26.5					87	111	48	90	92	174	205	30	7	78	14	50	825	.45
		Port	177	165	170	57	18	83.25	31.25	10	31.18	27.5					78		50	94	85		210			94			830	
7	4–19	St'b'd	170	233	235	90	28	92.5	42	13	39.12	26.5					87	112	49	86	86	177	270	38	7	98	17	50	900	.2
		Port	167	214.5	210	75	24	96	39.5	13.25	38.9	27.5					79		50	94	98		215			96			925	
8	4–33	St'b'd	245	248	265	95	30	97.5	48	16.4	45.25	26.5					87	112	49	95	86	175.5	275	50	8	100	20	55	950	.3
		Port	240	228	220	80	27	106.5	41.5	14.7	42.33	28					80		50	82	104		300						940	
9	4–52	St'b'd	275	248	265	105	34	117.5	55	20.6	54.36	26.5					87	116	48	100	102	183.5	305	68	60	105	20	55	1,600	.6
		Port	275	248	265	98	34	128.75	50	19.4	52.75	27.5					81		50	106	106		305						1,600	
10	5– 5	St'b'd	275	268	270	108	34	105	56	21	53.6	26.5					87	114	49	99	120	179	342	49	11	100	21	50	1,500	.45
		Port	271	258	265	100	34	123.5	51.25	19.75	52.9	27.5					81		50	90	104		350						1,500	
11	5–18	St'b'd	273	268	270	105	34	100	57.25	20.25	52.7	26.5					87	118	49	105	120	181	344	54	12	136	20	55	1,500	.5
		Port	275	258	265	100	34	125.75	49.5	18.9	51.7	26					81		50	122	117		310						1,500	


200 300 400 500 600 700 800 900 1000 1100 1200 1300 1400 1500
Indicated Horse Power
Speed in Knots
Mean Curve of speed and I.H.P. for runs south
Curve of speed and I.H.P. for runs south
Mean Curve (both engines) of speed and I.H.P.
Curve of speed and I.H.P. for runs north
Curve of speed and revolutions for runs south (with the tide)
Mean Curve of speed and revolutions
Curve of speed and revolutions for runs north (against tide)
U.S.S. DUBUQUE
Speed Revolution and Speed I.H.P. Curves constructed from data of progressive runs over measured mile course off
NEWPORT, R.I.
May 10-1905.
Plate 1.
Revolutions
110 120 130 140 150 160 170 180 190 200 210 220 230 240 250

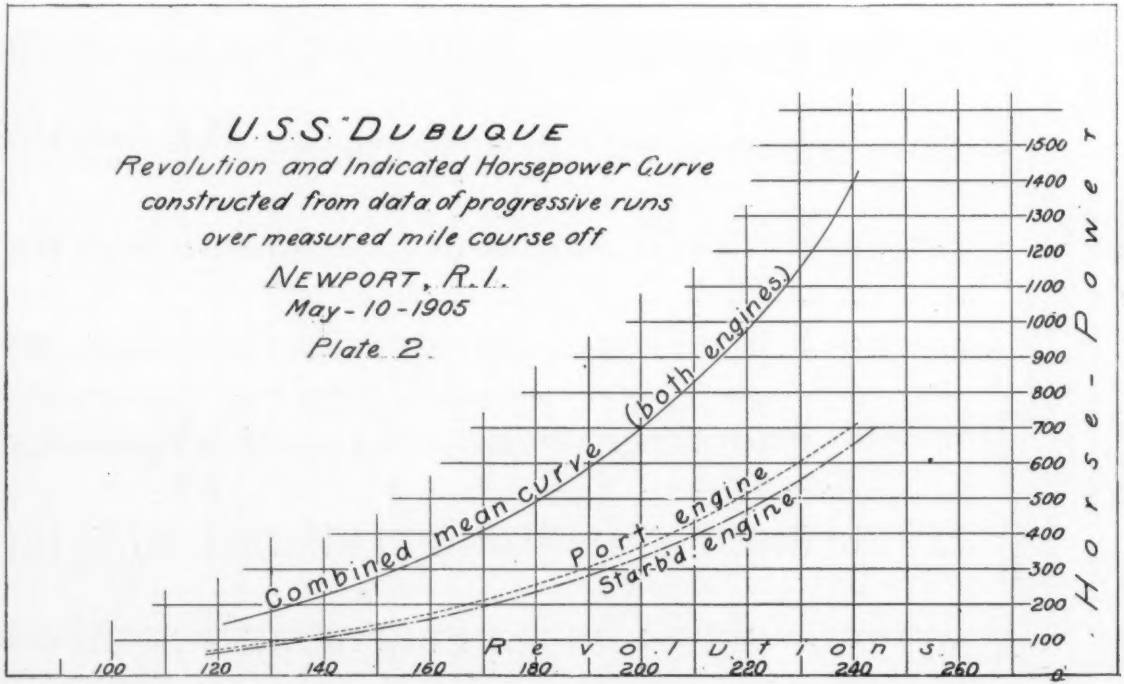

U.S.S. DUBUQUE
Revolution and Indicated Horsepower Curve
constructed from data of progressive runs
over measured mile course off
NEWPORT, R.I.
May-10-1905
Plate 2.
Combined mean curve (both engines)
Port engine
Starb'd. engine
Horse-Power
1500
1400
1300
1200
1100
1000
900
800
700
600
500
400
300
200
100
0
Revolutions
100
120
140
160
180
200
220
240
260

Table III.—U. S. S. *DUBUQUE.*

Data Deduced from Progressive Runs over Measured-Mile Course off Newport, R. I., May 10, 1905.

Speed in knots per hour.	Revolutions per minute.	Indicated horsepower.			Slip of propeller in per cent. of its own speed.	Constant R ÷ S.
		Starboard engine.	Port engine.	Total both engines.		
7	**112.5**	**52**	**60**	**112**	**7.72**	**16.000**
7.1	114.2	56	62	118	7.81	16.085
7.2	116	58	66	124	7.95	16.110
7.3	117.6	60	68	128	7.94	16.110
7.4	119.3	64	72	136	8.01	16.120
7.5	121	68	76	144	8.08	16.135
7.6	122.7	72	80	152	8.15	16.140
7.7	124.4	73	83	156	8.21	16.155
7.8	126.2	76	90	166	8.35	16.180
7.9	128	80	92	172	8.47	16.203
8	**129.8**	**85**	**95**	**180**	**8.59**	**16.22**
8.1	131.5	89	97	186	8.66	16.230
8.2	133.2	94	102	196	8.72	16.245
8.3	135	96	108	204	8.82	16.265
8.4	136.8	100	110	210	8.94	16.280
8.5	138.6	105	115	220	9.06	16.305
8.6	140.4	109	119	228	9.25	16.325
8.7	142.2	111	125	236	9.27	16.343
8.8	144.1	118	128	246	9.43	16.373
8.9	146	122	132	254	9.60	16.400
9	**147.7**	**127**	**137**	**264**	**9.64**	**16.410**
9.1	149.7	130	144	274	9.85	16.450
9.2	151.4	135	149	284	9.88	16.456
9.3	153.3	142	154	296	10.03	16.480
9.4	155.2	146	162	308	10.18	16.510
9.5	157	151	167	318	10.27	16.525
9.6	158.8	156	172	328	10.34	16.540
9.7	160.7	164	178	342	10.48	16.565
9.8	162.6	169	185	354	10.61	16.590
9.9	164.4	175	193	368	10.70	16.605
10	**166.4**	**185**	**199**	**384**	**10.89**	**16.640**
10.1	168.2	188	206	394	10.96	16.650
10.2	170.2	196	216	412	11.13	16.685
10.3	172.1	204	222	426	11.26	16.710
10.4	174.1	210	230	440	11.42	16.740
10.5	176.1	220	240	460	11.59	16.770
10.6	178.1	229	245	474	11.75	16.802
10.7	180.1	236	256	492	11.90	16.830
10.8	182.2	246	264	510	12.11	16.870
10.9	184.4	255	277	532	12.35	16.920
11	**186.3**	**264**	**284**	**548**	**12.44**	**16.935**
11.1	188.4	270	300	570	12.64	16.970
11.2	190.5	285	309	594	12.82	17.100
11.3	192.8	293	322	615	13.10	17.060
11.4	195	307	334	641	13.31	17.105
11.5	196.1	314	340	654	13.03	17.050
11.6	199.4	330	360	690	13.74	17.190
11.7	202	347	377	724	14.11	17.265
11.8	204	360	390	750	14.22	17.290
11.9	206.2	372	408	780	14.42	17.330
12	**208.6**	**388**	**424**	**812**	**14.70**	**17.380**
12.1	211	402	442	844	14.96	17.435
12.2	213.5	420	460	880	15.27	17.500
12.3	215.7	434	478	912	15.44	17.540
12.4	218.1	450	500	950	15.70	17.590
12.5	220.5	469	519	988	15.94	17.640
12.6	223	490	540	1,030	16.21	17.700
12.7	225.2	510	560	1,070	16.38	17.730
12.8	227.7	530	582	1,112	16.64	17.790
12.9	230.2	552	607	1,162	16.90	17.850
13	**232.6**	**576**	**630**	**1,206**	**17.09**	**17.890**
13.1	235.2	608	656	1,264	17.40	17.955
13.2	237.8	634	686	1,320	17.69	18.015
13.3	240.5	664	712	1,376	17.99	18.085

SMOKE OBSERVATIONS MADE DURING THE THREE MAXIMUM SPEED RUNS OF THE PROGRESSIVE-SPEED TRIAL.

Time smoke appeared.			Time smoke disappeared.			Duration of smoke.
Hrs.	*Min.*	*Sec.*	*Hrs.*	*Min.*	*Sec.*	*Sec.*
4	48	35	4	49	20	45
	50	35		51	30	55
	52	35		53	30	55
	54	30		55	20	50
	56	43		57	25	42
	58	43		59	30	47
5	00	54	5	1	50	56
	2	40		3	20	40
						—
						Average, 48

One raked and one door fired every two minutes.

THE FULL-SPEED FOUR-HOUR (PRELIMINARY) TRIAL.

On the 11th of May, at 8.45 A. M., at sea, the contract full-speed trial was begun and proceeded without mishap of any kind. For the purpose of avoiding a long and useless return run in addition to the trial run, the latter included a very gradual turn so as to bring the finish somewhere near the starting point.

The performance was in every respect satisfactory to the trial board. The ship steered well, turned in a small circle and was free from unusual vibrations. The vibrations of the main engines, never excessive, were greatest at 120 revolutions; at 220 revolutions it was almost *nil.*

All bearings ran cool; no water being used except on the thrust bearings as a precaution. Indicator cards were taken from all main cylinders every half hour simultaneously, and all engine data was recorded every fifteen minutes. Cards from the auxiliaries had been taken at various speeds previously.

The firing or stoking was systematically performed under the direction of a representative of the Babcock & Wilcox Co., at regular intervals of two minutes, thus:

Time.	Rake.	Fire.
4·36	2	1
38	3	2
40	1	3
42	2	1
44	3	2
46	1	3
48	2	1
50	3	2

The numbers indicate the furnace doors of each boiler; three furnace doors, one furnace to each boiler. Thus the fires of one-third of one furnace were being raked while fires of an equal portion of the other boiler were being coaled. A certain number of shovelfuls were thrown on at each firing, no hesitation or study on the part of the fireman being necessary. Considering the boilers, the method of firing and general excellence of the machinery, it is not surprising that the results were, for naval engines, so satisfactory.

As so much has been written about the smoke nuisance of certain of our ships, and for purposes of comparison, photographs were taken at regular intervals to include all conditions of firing during the four-hour run, showing the tops of the funnels and smoke issuing, which, by the way, was very slight. There was no smoke trail whatever.

PERFORMANCE ON PRELIMINARY TRIAL, MAY 11, 1905.

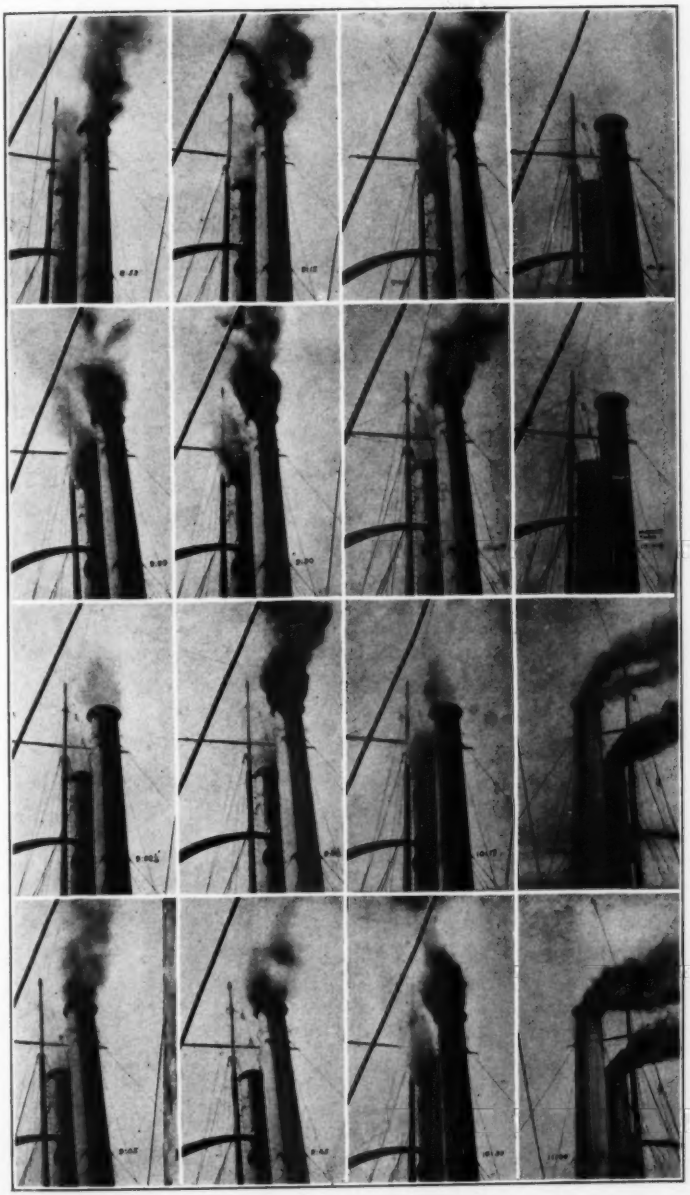
Trial began	8.45 A. M., May 11, 1905
Duration of trial (two hours seaward, a long turn and return), hours	4
Steam pressure at boilers (per gauge), pounds	240.8
H.P. steam chest, S. engine (per gauge), pounds	222.7
P. engine (per gauge), pounds	221.2
1st receiver, S. engine (absolute), pounds	94.2
P. engine (absolute), pounds	87
2d receiver, S. engine (absolute), pounds	29.4
P. engine (absolute), pounds	30

MEAN EFFECTIVE PRESSURES IN CYLINDERS IN POUNDS PER SQUARE INCH.

Main engines:	
Starboard	H.P., 105.38; I.P., 51.39; L.P., 17.39
Port	H.P., 113.87; I.P., 45.10; L.P., 17.01
Mean pressure in pounds per square inch on L.P. piston equivalent to aggregate M.E.P. on all pistons, starboard	48.41
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, port	46.82

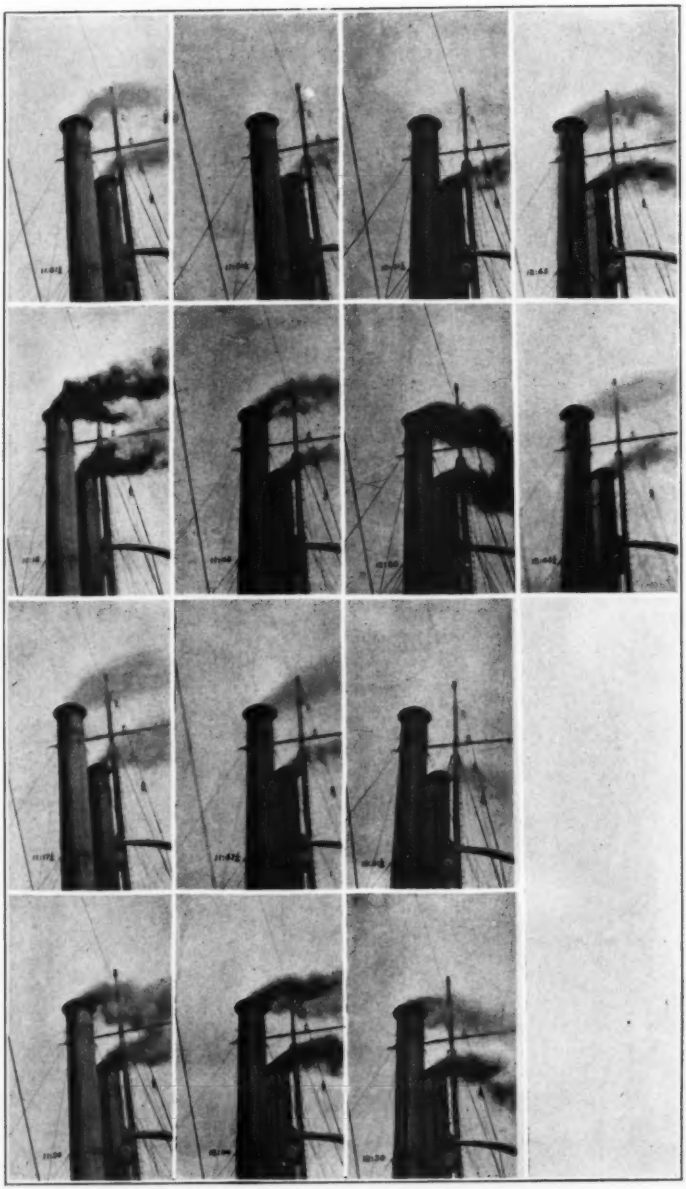
REVOLUTIONS PER MINUTE.

Main engines	starboard 227.73; port 231.79
Pumps, air	starboard, attached; port, attached.
Pumps, circulating	starboard 264.6; port 311
feed, double strokes per minute	46.5
auxiliary, double strokes per minute, water service	14
bilge	50



U. S. S. "DUBUQUE."

Photographs of Smoke taken during Official Trial.



U. S. S. "DUBUQUE."

Photographs of Smoke taken during Official Trial.

Blower engines	forward, 447.7; after, 451.1
Speed of ship, in knots per hour	12.90
Slip of propeller, in per cent. of its own speed— Based on mean pitch. St'b'd... 16.32 Port... 17.79 Based on final pitch. St'b'd... 16.32 Port... 17.79	
Air pressure in fireroom, in inches of water	.125
at base of stack, in inches of water	.5
Temperature at base of stack, degrees Fahrenheit	569.4
outside air, degrees Fahrenheit	77
fireroom, working level, degrees Fahrenheit	112

INDICATED HORSEPOWER.

Main engines:	
Starboard	H.P., 156.86; I.P., 224.17; L.P., 213.67; Total, 594.70
Port	H.P., 172.52; I.P., 200.24; L.P., 212.73; Total, 585.49
Air pump, starboard engine	Attached. Port engine Attached.
Circulating pump, starboard engine .62	Port engine .85
Feed pumps, main, one only installed	11.3
(Attached pump not used on trial.)	
Bilge pump	1
Water-service pumps on sanitary service	.5
Ventilating engines for ship, electric; for engine rooms, electric.	
Blower engines for forced draft in firerooms (2), total	4.2
Dynamo engines	21.4
Collective of all main engines	1,180.19
main engines, air, circulating and feed pumps	1,192.96
and auxiliary engines in operation during trial	1,220.06
all machinery during trial, per square foot of G.S.	12.228
H.S.	.293
Main engines, air, circulating and feed pumps, per square foot of G.S.	11.95
Main engines, air, circulating and feed pumps, per square foot of H.S.	.287

COAL.

Kind and quality	Pocahontas hand-picked, excellent.
Pounds per hour	1,903.52
Pounds per hour, per I.H.P., collective, all machinery in operation	1.559
Pounds per hour, I.H.P., collective, main engines, air, circulating and feed pumps	1.596
Pounds per hour, per square foot of G.S.	19.08
H.S.	.4577
Cooling surface, square feet per I.H.P.	1.113
Heating surface, square feet per I.H.P.	3.409

THE SMOKE QUESTION.

RELATION BETWEEN SMOKE DEVELOPMENT AND UTILIZATION OF FUELS, AND THE WAYS AND MEANS OF SMOKE ABATEMENT DURING FIRING.

By F. Haier, Hamburg.*

Translated from the "Zeitschrift des Vereines Deutscher Ingenieur" by William Wachsmann, Associate Member.

Gentlemen, respecting my presentation I must remark that, owing to the extent of the field in question and the difficulties met with, a complete and exhaustive treatment cannot be considered. In the following I will restrict myself principally to the field of steam-boiler furnaces, and of this only so much shall be considered as will be necessary to indicate the way in which, in my opinion, a healthy progress must be directed.

In the course of this discussion I will show how the *Society for the management of furnaces and controlling of smoke*,† formed in Hamburg in 1902, took hold of this problem, and how improvement in this field is to be brought about.

As to the annoyances, burdens and damages produced by smoke, further comment is unnecessary. We all concede that smoke is no agreeable accompaniment to firing. But, on the other hand, this much is also established, that as long as we

*Lecture delivered before the Hamburg and Berlin branches of the Society of German Engineers.

†The circumstances surrounding the formation of this Society were brought about by the contemplated action of the Hamburg municipal authorities to proceed against the smoke and soot evil of industrial furnaces. Against the issuing of such rules, however, vigorous opinions became manifest in manufacturing circles. The committee on industry of the Chamber of Commerce expressed themselves in the senate to the effect that improvement of the conditions in question could be more certain and more easily obtained by way of self help—the industrial circles interested coming together and forming a society along the lines of the Boiler Guardian Society, and declaring that they were ready to form such a society. Thereupon the contemplated municipal measures were dropped to await the activity of this association, which, in the fall of 1902, after the completion of certain preliminaries, took up its labors, and for its object had "the procuring of the most perfect and smokeless combustion with the greatest possible utilization of fuel in the installations of its members." According to the by-laws of the Society this object was particluarly to be brought about by:

"1. Regular inspection of boiler plants, including the supervision of their service and proposals to remedy any defects that might be found.

are compelled to burn coal complete prevention of smoke is out of the question. We must rather be satisfied to restrict this evil within certain practically-attainable limits. Furthermore, while industrial furnaces are generally held responsible for these annoyances, there is no doubt that in many cities during the winter months the domestic furnaces are no less the transgressors; but to get at these is more difficult than in the case of industrial furnaces.

The difficulties of the question can be seen from the attempts made for relief by means of laws. In no instance up to the present time have such laws produced any satisfactory results. Recalling the case of England, for example, where this question, due to the early development of its industries, first came up, no considerable progress has as yet been made despite the existence, for decades, of a smoke-prohibition law. Even with us, in Germany, attempts to progress by way of municipal measures have also been made at different places. If success has not attended these efforts to adjust the conflicting interests, then, in my opinion, it is because, in the case of this complicated problem, many facts remain to be established.

There is hardly any branch of engineering in which so many contradictory views are still held and, in spite of the extraordinary inventive ability displayed, no such extreme methods of practice are to be found as in this field; this more than anything else, it seems, would show that much which is essential yet remains to be made clear.

Before entering upon the discussion of the ways and means

"2. Dealing with all questions, such as the management of furnaces, utilization of fuels, smoke production, the plant and the modification of furnaces, heat value of fuels and questions relating thereto, the giving of opinions concerning all these, also the supervision of plants in the course of construction and those to be rebuilt, etc.

"3. The education of firemen.

"4. The collection of information obtained from the actual operation of such plants, particularly the testing of special firing contrivances, and the dissemination of the same among its members."

The members who, besides the payment of annual dues, pay a yearly fee for each furnace plant and fees for special work (evaporating tests, opinions, consultations), may demand that their furnace plants be regularly examined at least three times a year. They pledge themselves to follow the instructions of the Society's officers and to notify its technical staff of any new installations or rebuilding of existing ones which may be contemplated.

For the development of the Society and its method of procedure see its first annual report, 1902 to 1903 (published by Boysen & Maasch, Hamburg).

of smoke abatement, there is one point I would like to examine more closely which, above all, is of vital interest to industry and is also of fundamental importance in judging the whole question, viz: the relation between smoke development and the utilization of fuels. Upon this question the views held in engineering circles are still far apart and the clearing up of this in the beginning would, in itself, be an appropriate contribution toward a healthy progress.

I. RELATION BETWEEN SMOKE DEVELOPMENT AND THE UTILIZATION OF FUELS.

In firing coal rich in gas, therefore inclined to make smoke, good heat utilization is not always combined with light smoke or smokeless working. Both *might* happen; but there are cases where, in spite of a smokeless stack, only a relatively small fraction of the heat contained in the coal is utilized while, on the other hand, even with considerable smoke making an extreme utilization of the coal can occur.

To explain these conditions more fully, I consider it proper, first of all, to recall the losses which take place during combustion.

When coal is burnt in a furnace a small quantity will always fall through the grate, likewise a small quantity will be inclosed by the coke. This coal is therefore withdrawn from combustion. Only in very rare cases are the flame-producing gases given off by the coal entirely consumed; on the contrary, these gases escape, more or less, without being consumed; generally a partial decomposition takes place during which carbon is separated in the form of soot. Consequently through these gases and vapors, partly visible and partly invisible, as well as through the soot, a certain quantity of heat is lost.

Of the actual heat developed, only a part is transferred to the heating surfaces as *useful*. A considerable portion escapes with the hot gases as free heat, therefore invisible, through the smoke pipe, while a further portion is lost by useless

radiation from the boiler and brickwork or by conduction to surroundings.

For the conditions here outlined only the two losses through the stack come into consideration, namely, those due to the free heat in the waste gases and those due to imperfect combustion.

The first-mentioned loss is found by ascertaining the quantity of the waste gases, their temperature, specific heat and composition; also the composition of the fuel, in order that the amount of gas generated per kilogram may be calculated.

While, in general, the ascertaining of this loss can be accomplished without any special difficulty, unfortunately there has been lacking a method sufficiently accurate for the direct determination of the loss through imperfect combustion, for which reason this determination has always been avoided. This also accounts for the still very incomplete knowledge as to the relative amount of this heat.

As the heat lost through radiation likewise cannot be directly determined, it is combined with the loss due to imperfect combustion and designated as the residue in the heat balance.

Of all losses in a furnace, the one most apparent and disagreeably felt is that due to imperfect combustion. This, no doubt, is also the reason why those who are not better informed as to the conditions have given this loss the greater prominence.

Conditions being equal, there is no question but that, from an economical standpoint, imperfect combustion is more detrimental than a perfect one. The question is, how far is it possible to bring about perfect combustion and at the same time restrict the loss of free heat in the waste gases, and what importance is to be attached to the heat lost, economically considered, during practical working by imperfect combustion?

To answer this it will be necessary to get an idea of the loss of the waste heat and the circumstances influencing the same.

The loss of the waste heat naturally increases with the quantity and temperature of the waste gases. Therefore it becomes greater the greater the excess of air during combustion, since

thereby the quantity of the hot gases naturally increases as well as their temperature, due to the conditions of heat conduction. The latter, moreover, becomes greater the more the boiler is forced.

Considering that 1 kg. of coal requires 7 to 9 cbm. of air, provided the process of combustion could be carried out with a quantity corresponding to its chemical composition, it is at once manifest that we have to deal with large gas volumes which in practice become still greater, as the above quantity is insufficient; it requires rather a multiple of this, depending upon the design of the furnace and the method of working adopted by the fireman. Moreover, for practical reasons, the temperature of waste gases below 180 to 200 degrees C. can no longer be utilized in a boiler.

If, for instance, combustion is effected with twice the volume of air, theoretically required (and even this does not indicate an unfavorable method of working) we have 16 cbm. of gas for 1 kg. of coal of average composition, supposing this reduced to a temperature of 0 degree C. and barometric pressure of 760 mm.

With a temperature of 250 degrees C. for the waste heat—which corresponds to moderate forcing for this method of working—there is in the heat lost about 1,180 T.U. for 1 kg. of coal, or, taking the heat value of the coal at 7,000 T.U., about 17 per cent. of the available heat.

This loss, entirely invisible to the eye, increases approximately in a direct ratio to a multiple of the minimum theoretical quantity of air and in direct ratio to the waste temperature. Cases where it reaches 30 per cent. are not in the least rare.

At any rate, from this it follows that the loss in the utilization of fuels is of considerable importance.

In opposition to this, what are the facts with regard to the loss through imperfect combustion?

As has already been said, the direct determination of this is very difficult. The only series of tests for its determination, based on a method of exact chemical measurement, date from

the years 1879 to 1881, and were carried out by Bunte in the heat-test station at Munich.*

The main results of these tests were, that from waste gases containing about 8 to 10 per cent. of carbonic acid, therefore containing about 100 per cent. excess of air, very nearly perfect combustion could, in most cases, be obtained. A loss due to imperfect combustion of over 5 per cent. of the heat value of the coal occurred, in general, only in tests where the conditions of combustion were intentionally made unfavorable.

Furthermore, these tests show that imperfect combustion is more easily avoided the greater the excess of air, within the practical limits, but with it the waste heat also becomes greater, and that, generally, a considerable loss through imperfect combustion is only to be feared when the air supply is very limited.

Therefore, even though the latter loss can become quite considerable, the same importance is not to be attached to it for the utilization of fuels as that to the loss in waste heat, since according to the foregoing the loss through imperfect combustion moves, as a rule, contrary to that of the waste heat. As the loss through imperfect combustion, to any appreciable extent, generally results from an insufficency of air, a comparatively small loss of waste heat is connected therewith. If the obtaining of perfect combustion is conditional upon a larger supply of air, then the loss of the waste heat will become greater and the resulting gain is once more, at least partly, lost.

Especially on account of this circumstance technical circles, until recently, were inclined to attach less importance to the loss through imperfect combustion than it deserved.†

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* See H. Bunte, "Zeitsch, d. v. d. I.," 1900, p. 674; also "Bayerische Industrie and Gewerbeblatt," 1879 to 1881, H. Bunte: Report of the heat-test station, Munich.

† Relative to this, see also the discussion upon the test series given in Table V, where, despite an increasing loss through imperfect combustion. the utilization is improved in consequence of the simultaneous drop in the loss of the waste heat. In addition to this—as a further reason for unduly underrating the loss through imperfect combustion—is the fact that, according to the usual method of calculating the waste-heat loss, in the case of very imperfect combustion, it becomes too great, and with that the increase of the residue loss becomes too small compared with the residue loss obtained for perfect combustion. See the discussion on p. 645 at the conclusion of note.

As to the conditions actually obtained, a series of trials which I carried out on a two-furnace boiler of standard make and which are to be discussed in the following, may furnish some information. The heating surface of this boiler was 75 square meters.

A clue to the amount of loss through imperfect combustion can be gained from the following method without the aid of an exact chemical analysis of the waste gases: Perfect combustion can be produced in a boiler by giving a proper supply of air and proper strength of draft; and by tests thus carried out, find the loss through radiation and conduction of the plant, from the residue in the heat balance, for the conditions of pressure in question.

Since this loss, with approximately equal pressure conditions on the boiler but different methods of working, will hardly vary, conclusions may be drawn—based upon a determination of the increase of the residue in the heat balance—through later tests to be made on the same plant, as to the loss due to imperfect combustion.*

The results of the tests carried out in this manner are given in Table I.

The construction and location of the boiler, which was in continuous service during the tests, can be seen in Figs. 1 to 3.

The four tests are not arranged in the order in which they were carried out, but according to the quantity of smoke developed and the quantity of air supplied. They were made between May 12 and 20, 1903, and the three designated I, II and III were each made twice under exactly the same conditions. The results of these duplicate tests agree with sufficient accuracy with one another. The duration of two tests was for a period of 9 and 9½ hours, respectively, and for all others, 10 hours.

The tests were made by taking accurate observations of all the conditions coming into consideration. The boilers for all the tests were attended by a fireman instructor.

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* Relative to this, see also p. 645, conclusion of note.

XL

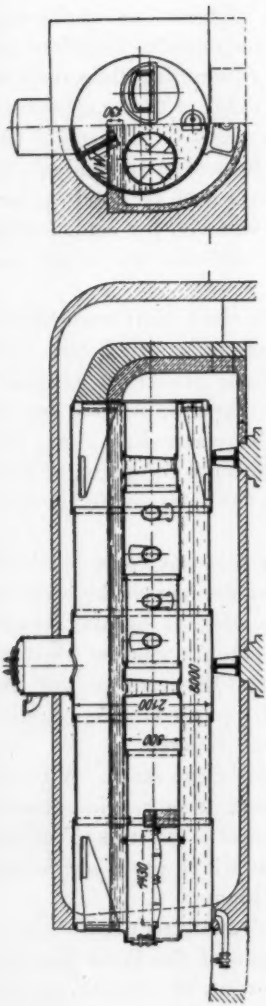
Table I.

		IV	
Heating surface of boiler, square meter	78.5		
Grate surface of boiler, square meter	2.28		
Ratio of grate surface to heating surface	1 : 34		
		May 20, 1903.	May 19
Duration of trial, hours		9	1
Fuel: English coal, West Hartley kind, weight of coal used during trial, kilograms		2,250	2,024
per hour, kilograms		250	202
square meter of grate surface, kilograms		109.6	88
heating surface, kilograms		3.18	2
Ashes, total, kilograms		190	240
in coal burnt, per cent		8.45	11
combustible consumed (carbon), per cent		22.6	25
Feed water: total evaporation, kilograms		12,192	13,078
evaporation per hour, kilograms		1,355	1,308
square meter, heating surface, kilograms		17.26	16
average temperature, degrees centigrade		12 4	12
Steam: steam pressure, atmospheres		6.64	6
average heat of evaporation, T. U.		645.2	644
Gas analysis: (a) at end of furnaces, per cent.		**14.6**	13
Carbonic acid, CO_2, average of 3-minute tests, per cent		14.45	13
15-minute tests, per cent		2.6	3
Oxygen, O average of 15-minute tests, per cent		17.05	17
Total of CO_2 and O, per cent		.....	.....
Excess of air (b) at end of boiler: carbonic acid, CO_2, average of 3-minute tests, per cent		9.3	8
15-minute tests, per cent		9.3	9
Carbonic oxygen, O, average of 15-minute tests, per cent		8.4	9
Excess of air, per cent		.....	.....
Temperature, degrees centigrade		273	230
of air for combustion, degrees centigrade		21	24
Draft pressure, at the boiler end, millimeters of water		5.4	4
Evaporation: (a) Water evaporated, per kilogram of fuel, kilograms		5.42	6
(b) Equivalent evaporation from and at 100 degrees centigrade per kilogram of coal from water at 0 degree centigrade, kilograms		**5.49**	6

Heat balance, see note page 641.	T. U.	Per cent.	T.U.
Utilized for making steam	3,497	54 4	4,165
Lost: (a) due to incomplete combustion of carbon	154	2.4	244
(b) due to heat carried away in the chimney gases	1,131	17.6	983
(c) other losses due to conduction, radiation, soot, unconsumed gases, etc	1,651	**25.6**	1,191
Total = heat value of fuel	6,433	...	6,583

Table I.

IV		I					III					II.				
May 20, 1903.		May 12, 1903.		May 13, 1903.		Average.	May 16, 1903.		May 19, 1903.		Average.	May 14, 1903.		May 15, 1903.		Average.
9		10		10		20	9½		10		19½	10		10		20
2,250		2,024		1,997		4,021	1,885		1,893		3,778	2,171		2,348		4,519
250		202		200		201	198		189		193.5	217		235		226
109.6		88.6		87.8		88.2	86.8		83		84 9	95.3		103		99.2
3.18		2.58		2.55		2.57	2.52		2.41		2.47	2.77		3		2.88
190		240		251		491	177		160.5		337.5	264		247		511
8.45		11.9		12.5		12.2	9.4		8.5		8.95	12.2		10.5		11.3
22.6		25.4		21.1		23.3	29		20.7		24 9	26.2		30.9		28.6
12,192		13,078		13,026		26,104	13,188		13,224		26,412	13,620		14,800		28,420
1,355		1,308		1,303		1,305.5	1,388		1,322		1,355	1,362		1,480		1,421
17.26		16.67		16.60		16.64	17.68		16.84		17.26	17.35		18,85		18.1
12 4		12.9		12.7		12.8	13		12.5		12.75	13		13		13
6.64		6.70		6.77		6.74	6.75		6.76		6.76	6.76		6.8		6.78
645.2		644.8		645.1		645	644.8		645.3		645	644.8		644.9		644.85
14.6		13.65		14.36		**14**	13.3		13.5		**13.4**	9.21		9.05		**9.13**
14.45		13.95		14.4		14.17	13.45		13.55		13.5	9.35		9.2		9.28
2.6		3.85		3.55		3.7	5.3		5.05		5.17	9.6		10.35		9 97
17.05		17.80		17.95		17.87	18.75		18.6		18.67	18.95		19.55		19.25
.....		.....		.....		.....	37		35		36	91		102		96
9.3		8.77		8.46		8.62	9.4		9.24		9 32	6.4		6.7		6.55
9.3		9.3		8.5		8.9	9.1		9.1		9.1	6 4		6.6		6.5
8.4		9.5		10.4		9.95	9.8		9.8		9.8	12.8		12.8		12.8
.....		.....		.....		.....	93		94		93.5	183		171		177
273		230		236		233	271		290		281	320		326		323
21		24		22		23	21		23		22	22		23		22.5
5.4		4.7		4.6		4.65	5.5		5.8		5.65	12.6		12.6		12.6
5.42		6.46		6.52		6.49	7		6.99		6.99	6.27		6.3		6.29
5.49		6.54		6.60		**6.57**	7.08		7.08		**7.08**	6.35		6.38		**6.37**
T. U.	Per cent.	T.U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.
1,497	54 4	4,165	63.3	4,206	63.65	4,186	63.5	4,514	68.5	4,511	70	4,513	69.25	4,043	59.9	4,063	62.1	4,053	61
154	2.4	244	3.7	215	3.25	229	3.5	220	3.3	142	2.2	181	2.75	258	3.8	264	4	261	3.9
1,131	17.6	983	14.9	1,051	15.9	1,017	15.4	1,116	16.9	1,211	18.8	1,164	17.85	1,948	28.85	1,852	28.3	1,900	28.6
1,651	**25.6**	1,191	18.1	1,136	17.2	1,163	**17.6**	737	11.3	578	9	657	**10.15**	502	7.45	365	5.6	434	**6.5**
6,433	...	6,583	...	6,608	...	6,595	...	6,587	...	6,442	...	6,515	...	6,751	...	6,544	...	6,648	...



Figs. 1 and 2.

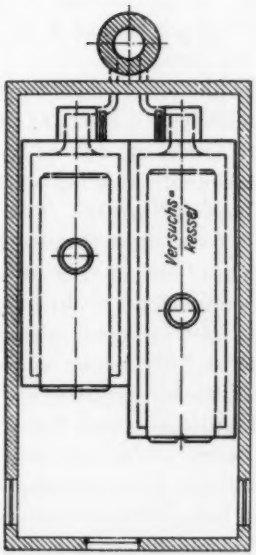


Fig. 3.

The corresponding smoke diagrams are shown in Fig. 4.

In tests IV and I combustion ensued with an average proportion of 14.6 and 14.0 per cent. of carbonic acid at the end of the furnace. As they were very incomplete, therefore no conclusion can be drawn from them concerning the excess of air without a knowledge of the percentage of the unconsumed gases;* but at any rate it was very small. In test III the average proportion of carbonic acid formed at the end of the furnace was 13.4 per cent., and the excess of air about 35 per cent.; while for test II this excess was 100 per cent. for an average proportion of 9.1 per cent. of carbonic acid at the end of the furnace.

In the latter test the entire available stack draft was utilized amounting to about 13 mm. water column at the boiler end. The tendency was to produce as little smoke as possible, consequently to bring about perfect combustion without regard to heat utilization. The coal was thrown mainly along the center and more toward the front portion of the grate, so that plenty of air could be admitted along the sides and in the rear for combustion of the gases.

In test III good utilization of the fuel was to be obtained with as little smoke as possible. The fireman worked with a considerably reduced air supply, therefore a weaker draft, amounting to only 6 mm. of water column against 13 mm. as before. The coal was thrown upon the front part of the grate and after burning was carefully pushed back, especial care being taken to keep the grate well covered.

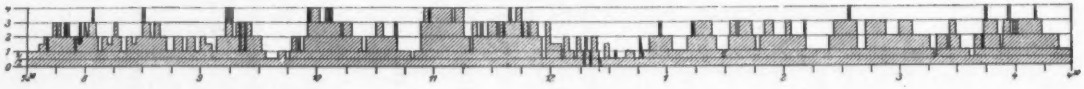
Test I was made with still less draft in the stack. Without any regard to the smoke, the best possible proportion of carbonic acid and the least possible excess of air was to be obtained. The coal was thrown upon the whole grate *uniformly* and the burning layer thereby kept thicker.

In test IV the excess of air was still more reduced.

The coal used was of English origin, of the West Hartley kind, containing about 27 to 29 per cent. of volatile matter,

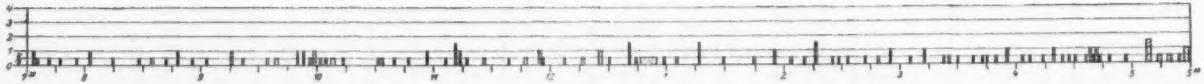
* In relation to this, see also p. 645, conclusion of note.

TEST IV, MAY 20, 1903.

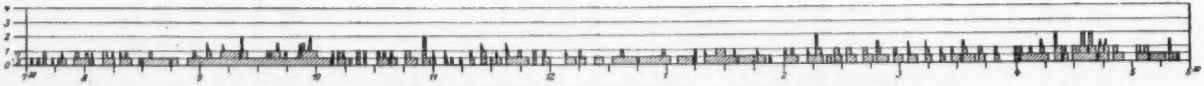


TEST I, MAY 13, 1903.

TEST III, MAY 19, 1903.



TEST II, MAY, 14, 1903.



KIND OF SMOKE.—0 = no smoke. 1 = light transparent smoke. 3 = dark gray smoke.
½ = trace of smoke. 2 = light gray smoke. 4 = black smoke.

Fig. 4.

therefore strongly inclined to smoke making. The average heat value was about 6,500 T. U.*

The hourly evaporation per square m. of heating surface was nearly the same on all days, averaging 16 to 18 kg., and can therefore be considered moderate. The coal consumption varied, according to the utilization desired, between 83 and 110 kg. per square m. of grate surface an hour.

VARIATION IN HEAT DISTRIBUTION, WITH THE AIR SUPPLY IN THE ORDINARY GRATE.

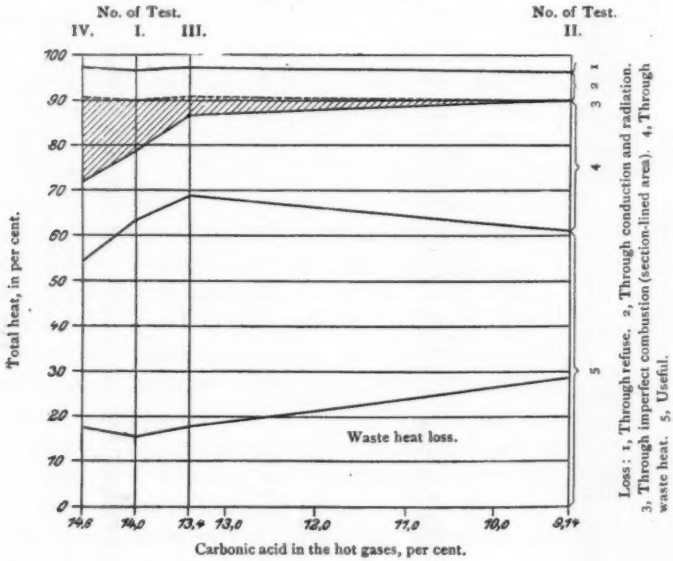


Fig. 5.

The graphical representation in Fig. 5 gives a very good idea of the results of the test. We see that the loss through imperfect combustion, which is represented by the shaded surface, can, indeed, obtain a not insignificant importance; but this first arises from conditions seldom occurring in practice, at least, not permanently. For operating under such condi-

*See Table III.

tions (especially as they existed for test IV, and also for test I, weak draft at the boiler end, at the same time maintaining a heavy and uniform thickness of coal) is rather exceptional in practice. Further, it can be seen that the utilization was best when moderately smoking, and by increasing the supply of air the smoke could be thoroughly checked, but then the *utilization of the fuel*, in consequence of the great increase in the loss of free heat in the waste gases, decreased *rapidly* and became still less than in the case of test I, where imperfect combustion had already set in to a considerable degree. In the latter it still amounted to 63.5 per cent., compared with 61 per cent. during test II, with practically no smoke. But for *very* imperfect combustion, during test IV,* it fell rapidly below this figure to 54.4, while during test III it rose to 69.25† per cent. with moderate smoke making.

NOTE REFERRED TO IN TABLE I.

Regarding the heat balance, there is this to be said: The heat values have been recalculated on the basis of the quantity of pure ash established during the various tests. With the original values the distribution of the heat would have been as found in Table II. With respect to this, see the foot note to Table V.

The analyses of the coal samples are given in Table III.

The loss of free heat in the waste gases was calculated according to the formula as laid down in the standard rules for testing boilers and engines, viz:

$$V = \left(0.32\,\frac{C}{0.536k} + 0.48\,\frac{9H + W}{100}\right)(T - t)\ \text{T. U.},\quad .\quad (1)$$

---

* That imperfect combustion existed in a high degree during this test is evident from the after ignition observed in the upper flue, which continued for about five minutes, and during which the temperature at the end of the boiler rose to about 500 degrees C, the average observed being about 273 degrees C. Interesting also is the comparison of the total percentages of the carbonic acid and oxygen during the four tests, which, according to table I, averaged 17.05, 17.87, 18.67 and 19.25, which figures likewise indicate imperfect combustion with progressively decreasing air suppy.

† With respect to these figures it must be stated that the conditions of utilization during all of the tests have somewhat suffered, due to the porous condition of the brickwork around the boiler. This could not be altered for the tests, but it really does not prejudice the insight into the conditions under discussion.

Table II.

Heat balance.	IV		I						III						II					
	T.U.	Per cent.	T U.	Per cent.	T.U.	Per cent.	T.U.	Per cent.	T.U.	Per cent.	T.U.	Per cent.	T.U.	Per cent.	T.U.	Per cent.	T.U.	Per cent.	T,U.	Per cent.
Utilized for making steam.....	3,497	56.5	4,165	63.6	4,206	62.9	4,186	63.25	4,514	69.4	4,511	70.6	4,513	70	4,043	61.2	4,063	65.1	4,053	63.15
Lost: (a) due to incomplete combustion of the carbon...	154	2 5	244	3.7	215	3.2	229	3.45	220	3 4	142	2.2	181	2.8	258	3.9	264	4.2	261	4.05
(b) Due to heat carried away in the chimney gases.....	1,090	17.6	978	14.9	1,062	15.9	1,020	15.4	1,102	17.	1,201	18.8	1,151	17.9	1,906	28.9	1,767	28.3	1,836	28.6
(c) Other losses due to conduction, radiation, soot, unconsumed gases, etc.....	1,450	**23 4**	1,163	17.8	1,199	18	1,181	**7.19**	664	10.2	533	8.4	599	**9.3**	398	6.	149	2.4	274	**4.2**
Total=heat value of fuel..	6,191	...	6,550	...	6,682	...	6,616	...	6,500	...	6,387	...	6,444	...	6,605	...	6,243	...	6,424	...

Table III.

Number of trial	IV	I		III		II		Average.	Variation.
Date of trial	May 20, 1903.	May 12, 1903.	May 13, 1903.	May 16, 1903	May 19, 1903.	May 14, 1903	May 15, 1903.		
Carbon, per cent	65.57	69.12	69.70	68.42	67.50	69.27	66.03	.....	
Hydrogen, per cent	4.30	4.74	4.76	4 61	4 68	4.58	4.36	.....	
Oxygen and nitrogen. per cent	8.72	9.48	9.54	9.78	9.66	8.55	9 26	.....	
Sulphur, per cent	1.71	.84	.91	1.17	1.13	1.01	1.14	.....	
Ash, per cent	9.17	8.78	8.36	7.24	7 28	10.02	10 51	8.77	
Moisture, per cent	10.53	7.04	6.73	8.78	9 75	6.57	8.70	.....	
Volatile matter, per cent	27.12	28.9	27.6	28.45	26 81	27.55	26.9	.....	
Fixed carbon, per cent	53.18	55 28	57 31	55.53	56.16	55.86	53.89	.....	
Heat value of coal, T.U..	6,191	6,550	6,682	6,500	6,387	6,605	6,243	6,451	+3.58 p.c. −4.03 p c.
combustible, T.U	7,788	7,831	7,917	7,803	7,768	7,966	7,792	7,838	+1.63 p.c. − .89 p.c.
Percentage of ash in total coal consumed, per cent	6.96	8.36	9.29	6.13	6.57	8.19	6 65	7.33	.....
Heat value of combustibles, T.U	6,433	6,583	6,608	6,587	6,442	6,751	6,544	6,564	+2.85 p c. −2. p.c.

in which the part contained in the refuse and withheld from combustion has been deducted from the total carbon.

This formula was criticised by Eberle in the "Journal of the Bavarian Steam Boiler Inspection Society," of 1902, pp. 25 and 26, where he proves that it is only strictly valid in case the velocity of combustion varied proportionately to the amount of carbonic acid, while for a uniform velocity of combustion the calculation should be made according to the following formula

$$V = \frac{a}{n} \Sigma \left(\frac{T - t}{k} \right) + \frac{b}{n} \Sigma (T - t)$$

in which

$$a = 0.32 \frac{O}{0.536}, \; b = 0.48 \frac{9H + W}{100} \quad . \quad . \quad . \quad . \quad (2)$$

and n the number of simultaneous observations of the percentage of carbonic acid and temperature of the waste gases.

Eberle concludes from his deductions that the calculation of the loss of the waste heat from the temperature and the percentage of carbonic acid of the waste gases can only be regarded as reliable if the velocity of combustion is uniform, and then the calculation should be according to formula (2).

For the present tests the values found from both formulae are given in Table IV.

Table IV.

Number of trial	IV	I		III		II	
Date of trial	May 20, 1903.	May 12, 1903.	May 13, 1903.	May 16, 1903.	May 19. 1903.	May 14, 1903.	May 15, 1903.
Loss of heat in waste gases, T.U. :							
According to formula (1)..	1,128	983	1,051	1,116	1,211	1,948	1,852
According to formula (2)..	1,172	1,015	1,100	1,153	1,233	2,078	1,936
Difference:							
T. U	44	28	49	37	22	130	84
In per cent. of the heat value of the coal	0.7	0.42	0.74	0.56	0.34	1.93	1.27

As is evident, the variations are slight, and for the large loss of 30 per cent. in waste heat during test II attain values of

over 1 per cent., viz: 1.27 and 1.93 per cent., respectively, of the heat value of the coal.

Now, before heat tests can have any claim to accuracy, without doubt one's efforts, for other good reasons, would be to establish equal conditions. In the present case it was endeavored to attain this if possible where, of course, slight fluctuations in the duty, due to the respective conditions of management, could not be entirely avoided. Even if complete uniform duty could be established and all other conditions coming into consideration remain invariable, so far as this is possible, still approximately equal velocity of combustion could only be obtained in furnaces by continuous stoking (*e. g.* by mechanical stoking, inclined grates and similar constructions). But then, at the same time, the influence of the fluctuations in carbonic acid and of the temperature of the waste gases would be so slight that any variation obtained from the two formulae for the waste heat loss will hardly exceed the limits of accuracy of the determination.

In the ordinary grate, on the contrary, when periodically stoked and when using fuels very rich in gas, it is generally impossible, though all other conditions remain the same, to obtain a velocity of combustion which does not vary. This will rather vary from one period of stoking to another in such a manner that it will increase rapidly, immediately after stoking, with the ignited and freshly-thrown on coal, and will then again gradually decrease with the diminishing gas abstraction. That this is a fact is already evident from the amount of air required for such furnaces, in case perfect combustion is to be obtained, being generally greater after than before stoking.

The fluctuation, being conditioned by the more intense combustion of the volatile matter in the fuel, will become greater the more gas the coal contains and the more rapid the abstraction of gas.

If no special regulation of the air supply takes place, the percentage of carbonic acid in the heat gases will be found considerably greater during the abstraction of gas than after the completion of the same, so that, as a matter of fact, it

varies in a similar manner as the velocity of combustion, being calculated according to formula (1), therefore in this case at least just as much qualified as formula (2).

On the other hand, if less fluctuation in the percentage of carbonic acid is obtained by a proper regulation of the air supply, then the variation in the calculation according to both formulae will be insignificant.

Hence if proper care is taken to maintain nearly uniform conditions during a test, the determination of the waste heat loss according to formula (1), therefore based upon average values, will be completely within the limits of accuracy even for the ordinary grate,—naturally presupposing that carbonic acid tests are frequently made.

Concerning the waste heat losses and the residue not accounted for in the heat balance, it may be said, in the case of very imperfect combustion (as for example prevailed during tests IV and I), that the waste heat loss obtained, according to the usual method of calculating, is too large; the residue loss, therefore, becomes too small, so that from a comparison of the residue losses thus obtained the loss through imperfect combustion, as represented in Fig. 5, will be too small rather than too large. For such conditions the excess of air and volume of the gases of combustion obtained will likewise be too large if calculated according to the usual method for perfect combustion.

If, in the case of imperfect combustion, the combustible in the waste gases be determined according to the method practiced by Bunte for heat tests in the heat-test station at Muenich (a method quite recently improved by him), whereby it can be assumed, without any appreciable error, that all carbon as carbon oxide, and all hydrogen, exist in an independent state, a more accurate value for the waste heat loss can be calculated as follows:

Let C represent the fraction of the carbon contained in 1 kg. of coal to be burnt. Further, let c_1 be the loss of carbon per kg. in the refuse, c_2 the corresponding loss in unconsumed gases, and c_3 that of the soot, and let k_1 represent the percentage

of carbonic acid, k_2 the percentage of carbon oxide, h the percentage of hydrogen, and r the grammes of soot found in 1 cbm. of dry gas. Since during the combustion of 1 kg. of carbon in pure oxygen $\frac{1}{0.536}$ cbm. of carbonic acid is generated (referred to 0 degree C. and 760 mm. barometric pressure), then, if V_g be the volume of dry gas generated from 1 kg. of coal, there results the following relation for the same:

$$\frac{C - c_1 - c_2 - c_3}{0.536} : V_g = k_1 : 100$$

or

$$V_g = 100\left(\frac{C - c_1 - c_2 - c_3}{0.536 k_1}\right) \quad . \quad . \quad . \quad . \quad (1)$$

In this expression c_1 is known. If R kg. of refuse with v per cent. of combustible were found after the combustion of B kg. of coal, we have

$$c_1 = \frac{v}{100}\frac{R}{B} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

For c_2 and c_3, we have the relations

$$\frac{c_2}{0.536} = \frac{k_2}{100} V_g \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

and

$$c_3 = \frac{r}{1{,}000} V_g \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The two latter expressions substituted in formula (1) give

$$C - c_1 - 0.536\frac{k_2}{100} V_g - \frac{r}{1{,}000} V_g = 0.536\frac{k_1}{100} V_g$$

or

$$V_g = \frac{C - c_1}{0.536\left(\frac{k_1}{100} + \frac{k_2}{100}\right) + \frac{r}{1{,}000}} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Further, let H be the fraction of hydrogen, and W the fraction of moisture per kg. of coal, and if H_1 is the loss of

hydrogen in kg. referred to 1 kg. of coal, then the steam generated from 1 kg. of fuel will be

$$G_w = [9(H - H_1) + W] \text{kg.} \quad . \quad . \quad . \quad . \quad (6)$$

in which

$$H_1 = 0.0896 \frac{h}{100} V_g \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Then from the specific heat c_g per cbm. of gas and c_w per kg. of steam the waste heat loss found will be

$$V_1 = \{c_g \frac{C - c_1}{0.536\left(\frac{k_1 + k_2}{100}\right) + \frac{r}{1{,}000}} + c_w [9(H - H_1) + W]\} (T - t) \text{ T. U.}, \quad . \quad . \quad (8)$$

in which

$$H_1 = 0.0896 \frac{h}{100} V_g$$

On the other hand, the loss through imperfect combustion found will be

$$V_2 = \left[\frac{k_2 + h}{100} V_g \cdot 3{,}000 + r V_g \times 8.1\right] \text{T. U.} \quad . \quad (9)$$

as the heat value of 1 cbm. of carbonic oxide as well as of 1 cbm. of hydrogen can be taken at about 3,000 T. U. and that of 1 gramme of soot at 8.1 T. U.

In Tables V and VI are compiled the results of a further series of tests, which are also very instructive.

These tests were made on a battery of five 2-furnace flue boilers, containing a total of 470 square meters of grate surface. The boilers were in continuous operation, and exhibited good performance conditions. The make and the arrangement of the boilers are shown in Figs. 6 and 7.

From a thorough examination of the waste gases, it was determined that the boilers were operated with too large an excess of air, in consequence of which the loss in free heat in the waste gases amounted to from 28 to 29 per cent. of the

Table V.

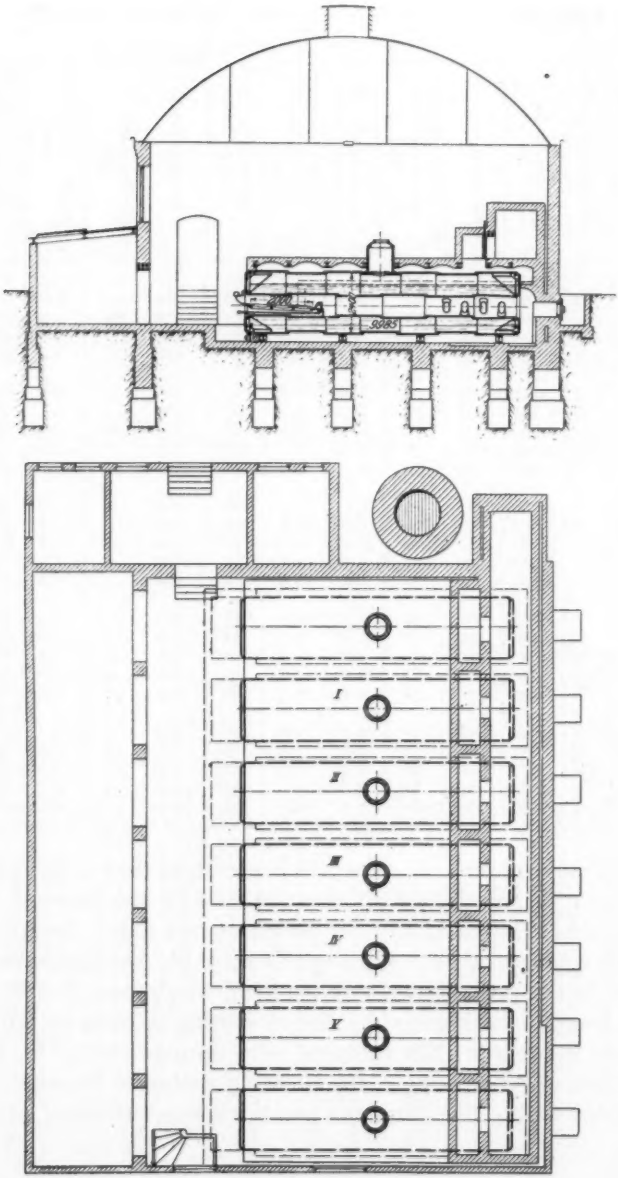
Boiler	I	II, III	IV, V	Trial 1, March 4, 1903.	Trial 2, March 5, 1903.	Average.	Trial 3, March 6. 1903.	Trial 4, March 7, 1903.	Average.	Trial 5, June 18, 1903.
Heating surface, in contact with water, sq. m.	103	84.7	99.1							
total sq. m.	120	110.2	123.2							
Grate surface, tests 1 to 4 sq. m.	2.98	2.8	2.72							
test 5 sq. m.	2.69	2.69	2.54							
Duration of trial, hours				10	10	20	10	10	20	10
Total heating surface, in contact with water, square meters				470.6	470.6	470.6	470.6	470.6	470.6	470.6
grate surface, square meters				14.02	14.02	14.02	14.02	14.02	14.02	13.15
Ratio grate surface to heating surface				1 : 33.6	1 : 33.6	1 : 33.6	1 : 33.6	1 : 33.6	1 : 33.6	1 : 35.8
Fuel: English coal, Real West Hartley main:										
Total coal burned, kilograms				10,615	10,562	21,177	9,532	10,728	20,260	11,135
per hour, kilograms				1,062	1,056	1,059	953	1,073	1,013	1,114
square meter of grate surface				76	75	75.5	68	77	72	84
heating surface				2.26	2.24	2.25	2.03	2.28	2.15	2.36
Ashes, total, kilograms				938	943	1,881	775	839	1,614	919
in coal burned, per cent.				8.8	8 9	8.85	8.1	7.8	8	8 2
combustible (carbon), per cent.				21.1	14.8	18	25.5	22.7	24.1	16.3
Feed water: total evaporation, kilograms				79,791	80,705	160,496	79,903	88,952	168,855	88,178
evaporation per hour, kilograms				7,979	8,071	8,025	7,990	8,895	8,443	8,818
sq. m. of heating surface, per hour, kilo.				17	17.2	17.1	17	18.9	17 9	18.7
average temperature, degrees centigrade				57.1	67.6	62.3	65.3	69.1	67.2	58.1
Steam: pressure, boiler I, atmospheres				6.7	6.6	6 65	6.5	6.7	6.6	6.5
boilers II to V, atmospheres				11.6	11.6	11.6	11.65	11.55	11.6	11.7
average heat of evaporation, T. U.				605.8	595.2	600 5	597.4	593.8	595.6	604.7
Heat gases: carbonic acid at the end of boiler, per cent.				**8.7**	**8.5**	**8.6**	**12.6**	**12.7**	**12.6**	**10.5**
excess of air at end of boiler, per cent.				112	117	115	46	45	46	76
temperature at end of boiler, degrees centigrade				**350**	**353**	**352**	**309**	**318**	**314**	**356**
of the air, degrees centigrade				18	20	19	23	23	23	27
Draft pressure at end of boiler, millimeters of water column				8.3	7.2	7.8	2.3	2.1	2.2	4.9
Evaporation: (a) Water evaporated per kilogram of fuel, kilograms				7 52	7.64	7.58	8.38	8.29	8.33	7.92
(b) Equivalent evaporation from and at 100° C. per kilogram of coal from water at 0° C., kilograms				7.15	7.14	7 14	7.86	7.73	7.79	7.52
Heat balance.*				T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.
Utilized for making steam				4,556	67.1	4,547	67	4,552	**67.05**	5,006	72.1	4,923	71.4	4,964	**71.75**	4,789	**72.3**
Lost: (a) in the refuse				152	2.2	107	1.6	129	1.9	168	2.4	144	2.1	156	2.25	109	1 6
(b) due to heat carried away in the chimney gases				1,690	24.9	1,715	25.3	1,702	**25.1**	1,048	15.1	1,055	15.3	1,051	**15.2**	1,384	**20.9**
(c) remainder=conduction, radiation, soot, unconsumed gases, etc.				389	5.8	412	6.1	400	**5.95**	722	10.4	777	11.2	749	**10.8**	343	**5.2**
Total = heat value of fuel				6,787	...	6,781	...	6,784	...	6,944	...	6,899	...	6,921	...	6,625	...

* See note page 658.

Table VI.—DATA TAKEN AT THE DIFFERENT BOILERS.

Number of trial.	Number of boiler.	Coal consumed.				Per cent. of carbonic acid at the end of boiler.	Excess of air at the end of boiler.	Temperature.		Draft at the boiler end.	Loss in free heat carried away in the chimney.
		Total.	Per hour.	Per hour per one square meter of grate surface.	Per hour per one square meter of heating surface.			Chimney gases.	Air for combustion.		
		kilos.	*kilos.*	*kilos.*	*kilos.*	*pr. ct*	*pr. ct.*	° *C.*	° *C.*	*mm. W. C*	*pr. ct.*
Trial 1.	I	2,203	220	74	2.13	8.4	120	327	...	10.2	24.1
	II	2,300	230	82	2.72	9.4	96	364	...	7.6	24.2
	III	2,209	221	79	2.61	10	85	376	18	7.5	23.4
	IV	1,970	197	72	1.99	8.3	122	362	...	7.6	27
	V	1,933	193	71	1.95	7.4	150	321	...	8.4	26.2
Trial 2.	I	2,144	214	72	2.08	8.6	114	326	...	8.6	23.3
	II	2,181	218	78	2.58	9	105	364	...	6.1	23.9
	III	2,158	216	77	2.55	9.8	88	385	20	6.2	24.9
	IV	2,094	209	77	2.11	8	130	367	...	7.4	28.4
	V	1,985	199	73	2	7.2	156	322	...	7.8	26.5
Trial 3.	I	2,166	217	73	2.1	11.5	60	308	...	3.6	16.4
	II	1,776	178	64	2.1	12.9	43	307	...	3.4	14.7
	III	1,681	168	60	1.99	14.1	30	324	23	1.1	14.5
	IV	1,952	195	72	1.97	13.1	40	313	...	1.5	14.7
	V	1,957	196	72	1.97	11.6	59	292	...	1.8	15.3
Trial 4.	I	2,213	221	74	2.15	12.1	52	302	...	3	15.1
	II	2,440	244	87	2.88	13	42	332	...	2.9	15.8
	III	1,930	193	69	2.28	13.7	34	335	23	1.3	15.3
	IV	2,174	217	80	2.19	13.6	35	327	...	1.8	14.9
	V	1,971	197	72	1.99	11.1	66	295	...	1.5	15.7
Trial 5.	I	2,479	248	92	2.41	10.4	78	359	...	6.7	21.2
	II	2,057	206	77	2.43	11.2	65	370	...	2.8	20.5
	III	2,479	248	92	2.93	10.6	74	404	27	4.7	23.8
	IV	2,072	207	81	2.09	10	85	344	...	4.7	20.9
	V	2,048	205	80	2.07	10.1	83	301	...	5.7	17.9

heat value of the coal, for which reason these tests were undertaken. It was decided to ascertain how far the excess of air could be restricted, and thus determine how much the utilization could be raised. The large excess of air was partly due to the fact that the grates were too long, which made it difficult to keep the back covered. Before starting the tests the grates were shortened. The furnaces were equipped with the Kowitzke upper-air supply, the air being delivered through the bridge wall. The duration of each test was ten hours. Dur-



Figs. 6 and 7.

ing the first two days of the tests the boilers were attended by the regular firing force from the factory. The excess of air compared with the original examination had somewhat decreased (as the boilers were operated under a pretty strong draft; moreover, the firemen stoked badly and did not permit the fires to burn down uniformly), but the loss in free heat through the smoke stack, according to the heat balance, still amounted to about 25 per cent. The utilization of the fuel was about 67 per cent., and in the refuse was found 1.9 per cent., leaving 6 per cent. as a residue. Since the development of smoke was moderate, it may be assumed that the residue loss of 6 per cent. is mainly to be credited to conduction and radiation. The residue loss can be considered as moderate, which is to be attributed to the good condition of the brickwork and lagging and to the small amount of radiating surface exposed by the five adjacent boilers.

I wish to add that, for the purpose of determining the waste heat losses as accurately as possible, the percentages of carbonic acid and the temperatures of the gases were taken every three or four minutes, at the end of each boiler. This was also done in the case of the first series of tests.

For the purpose of establishing the conditions that would obtain, if the method of working was changed, it was decided to let two firemen instructors take charge of the installation on the third and fourth days. The method of working was to be changed in such manner that the average per centage of carbonic acid in the waste gases obtained on the first and second days of the trials was to be increased from 8.6 to 12 or 13 per cent., or the excess of air of about 115 per cent. obtained on the first two days was to be cut down to about 40 or 50 per cent. To obtain perfect combustion of the gases under these conditions (the coal used contained about 30 per cent. of volatile matter) it was found that the grates were still too large despite the fact that they had already been shortened. To bring about the reduced excess of air the boilers had to be operated under a very light draft, and, although the doors for the supply of secondary air were constantly kept open, the

formation of smoke increased considerably. The comparison of the heat balances is very instructive. As a result of cutting down the supply of air, and in connection with this the drop in the gas temperatures at the end of the boilers, the loss in free heat in the waste gases fell from 25.1 to 15.2 per cent. —very nearly 10 per cent.—but at the same time very imperfect combustion was obtained. In consequence of this the residue loss in the heat balance was increased from about 6 to 11 per cent., therefore about 5 per cent. of the heat value of the coal was lost due to a more intense smoke formation.* But against this loss there was a falling off in the loss of the waste heat of nearly 10 per cent., so that in spite of the formation of smoke, which was considerable, the factor of utilization increased from about 67 to about 72 per cent.

Of course, matters were not permitted to remain thus: after the conditions had been cleared up, the cast-iron bridge walls were shifted and the grates still further shortened.

About three months later a further series of tests were made under the same conditions. It was found that the boilers could now be operated with a stronger draft, so that, for an average of 10.5 per cent. of carbonic acid in the waste gases, there was no excessive smoke developed. In spite of the fact that the boilers were not in as clean a condition as during the former tests (the waste-gas temperatures were relatively higher), the utilization obtained amounted to 72.3 per cent. The loss in free heat in the waste gases was 20.9 per cent., and the loss in the refuse, as well as in the residue, were about the same as originally. These losses are given in the last column of Table V.

In Table IX are given the results obtained in *water-tube boilers* with lignite, a fuel that is extraordinarily rich in gas.

As is well known, these boilers often cause the formation of very dense smoke, particularly with fuels very rich in gas, and therefore, in case the furnace is not well designed, they can cause fairly large losses through imperfect combustion.

*In regard to this see the remarks at the conclusion of note on p. 641, according to which the loss in waste heat was really still less and the increase of the loss through imperfect combustion was correspondingly greater.

Table IX.

Group.	I.		II.		III.		IV.		
					Ordinary grate.		Step grate with secondary air supply.		
							Steam generated in		
	Ordinary grate.	Step grate with secondary air supply.	Ordinary grate.	Step grate with secondary air supply.	Lignite.	½ Ruhr coal. ½ Lignite.	One boiler.	Two boilers.	
Heating surface, square meters	160	160	153	153	235	235	200	2×200=400	2×200=400
Grate surface, square meters	3.182	2.98	3.50	2.84	4.95	4.95	5.06	2×5.06=10.1	2×3.57=7.14
Steam generated per square meter of heating surface, kilograms	11.0	13.0	19.4	16.5	12.5	14.9	15.1	7.23	7.53
Coal consumption per square meter of grate surface, kilograms	128	188	181	143	167	99	121	50.8	73.6
Carbonic acid, per cent	9.5	7.2	11.3	9.7	9.7	7.8	14.5	8.6	7.7
Temperature of the waste gases, °C	371	341	405	284	376	311	374	267	250
Heat value, T. U	5,442	4,121	5,425	5,768	4,934	6,740	5,180	5,180	5,040
Heat utilized, per cent	50.5	57.5	55.1	68.9	45.9	67.5	62.2	70.2	72.1
Waste heat loss, per cent	25.3	31.6	23.3	18.2	26.1	25.7	17.4	19.3	19.7
Ashes, per cent	7.9	7.7	0.9	4.4	2.0	1.0	0.4	0.3	0.2
Residue loss, per cent	**16.3**	**3.2**	**20.7**	**8.5**	**26.0**	**5.8**	**20.0**	**10.2**	**8.0**

The figures in the first three columns are taken from an article by Eberle and Reischle, entitled "The main results from an engineering standpoint of the activity of the Bavarian Association for the Inspection of Steam Boilers," published in the "Zeitsch. d. Vereines d. Ing.," 1903, page 1039. These figures are of such value and interest that they are reproduced here. In tests I and II, the point in question was to replace the ordinary grates under such boilers by step-grates having an upper-air supply, according to the Dürr system. The figures in the third column are derived from two tests, both of which were carried out with the ordinary grate. One test was made with lignite, and the other with a mixture of $\frac{2}{3}$ Ruhr coal and $\frac{1}{3}$ lignite—a fuel very much poorer in gas.

In the article mentioned, the following remarks were made concerning these tests:

"By comparing only the residue losses in this table—in which, besides the losses through radiation and conduction, are also contained the losses through soot and unconsumed gases, as well as possible errors in observation—it can be seen that, through the substitution of the step-grate with a secondary air supply for the ordinary grate in tests I and II, the combustion, as well as the heat utilization, was materially improved. The same thing was obtained in test III by using a mixture of $\frac{2}{3}$ Ruhr coal and $\frac{1}{3}$ lignite, instead of all lignite, as used in the first test. In this test the residue loss was reduced from 26 to 5.8 per cent."

Further it is stated:

"If water-tube boilers are to be used for burning fuels rich in gas, as, for example, lignite, then in selecting the furnace arrangement especial care must be taken that the combustion is complete in the fire space, for, after the hot gases have once entered the tubes, further combustion is almost stopped, due to the fact that the gas stream has considerably cooled off and split up."

Similar results are shown in column 4, Table IX. These results are derived from the heat balance of tests taken from the "Journal of the Bavarian Association for the Inspection of

Boilers," November 15, 1903. The tests were made on two water-tube boilers of the same size, equipped with step-grates and an upper-air supply. One test was made with one boiler in use, and the other test with two boilers in use, the steam generation being the same in both cases. The fuel used was Bohemian lignite. The performance in the first test was 15.1 kg. of water evaporated per square meter of heating surface per hour, and in the second test, 7.23 kg. and 7.53 kg., respectively. The utilization in the test with two boilers amounted to 70.2 and 72.1 per cent., respectively. The report on the test says:

"The result of the test with one boiler with twice the steam performance was materially less satisfactory, since only 62.2 per cent. of the heat was utilized. But the falling off in the utilization in this test is to be attributed solely to imperfect combustion of the coal. Despite the fact that all openings for the supply of air were kept open and the draft in the stack was of the best, it was impossible to supply the quantity of air necessary for perfect combustion, the result being that the carbonic acid averaged 14.5 per cent.—which might be considered satisfactory, but at the same time there was a considerable loss in the residue, amounting to 20 per cent., one-half of which is probably due to the unconsumed particles in the hot gases. This result could be foreseen by the heavy emission of smoke from the stack during the tests."

From this it can be seen that, when burning fuels very rich in gas in water-tube boilers, there may be considerable losses if the furnace is not a suitable one for the boiler, or if the grate consumption is too high.

Returning once more to the first series of trials (Table I), test II shows distinctly that it is not a very difficult matter to operate boilers (at least in the case of flue boilers) having ordinary grates with little smoke emission, even for coal very rich in gas and a grate consumption up to 100 kg., as long as *no value* is placed upon the utilization of the fuel, and that perfect combustion—therefore smokeless combustion—can also exist without good utilization of the coal.

It will be understood that, so far as the latter point does not enter into the question, care need only be taken to provide a supply of air sufficiently large in the combustion space, under a good draft, so that it reaches the gases before they have been cooled below the temperature necessary for their ignition.

Moreover, the initial temperature (at least between certain limits), with regard to the formation of smoke, is of considerable less influence than a lack of air after stoking, as is seen from test II, where, as a result of the large excess of air, the temperature in the fire space doubtless was at its lowest; nevertheless the formation of smoke was least, due to the fact that the gases quickly obtained the air necessary for combustion That it was not a question of smoke abatement in this case, but one of more perfect combustion, is shown by the loss in the residue.

The foregoing statement is further verified by the series of tests in Tables I and V, which show that the same importance is not to be attached to the loss through imperfect combustion, for the utilization of fuels, as that to the loss in free heat in the waste gases, even though imperfect combustion, in certain cases, can undoubtedly assume an importance not to be neglected.

As already mentioned, in the practical operation of boilers (especially *flue boilers*), such large losses through imperfect combustion, as were intentionally brought about in the first named series of tests, are exceptional. The reason for this is that, as a rule, smoke development usually occurs only for a short period after the stoking and working of the fires—and especially when the fireman permits the fires to get too low, then upon stoking throws in too much coal and imprudently stirs the fire. If in such a case, by working more carefully, not only a decrease in smoke formation occurs, but the utilization itself can also be considerably raised, then the latter is due to the fact that the waste heat loss becomes less.

With the method of firing—frequently met with—there arises, besides the temporary intense smoke development, generally a large excess of air, and with this a heavy loss in waste heat. The formation of smoke is of course brought

about through a lack of air, which only continues during the very rapid abstraction of the gas from the fuel. After the termination of the gas abstraction, the quickly formed vacant places on the grates produce a considerable excess of air, and in this way cause an increase in loss through waste heat.

The increasing of the utilization by adopting a better method of working is therefore not to be attributed—as the layman is inclined to accept—to the difference in loss through imperfect combustion, but is mainly due to the variation of the losses in free heat in the waste gases.

Therefore these tests and discussions show that the development of smoke is not, as a rule, accompanied by a large loss in fuel, as is generally supposed by those who are not familiar with the question. *Undoubtedly as a result of a lack of understanding of the principles involved in the management of furnaces, vast quantities of heat are lost, so that it well pays to give more consideration to this than has generally been the case.* The losses are not so much to be sought in the smoke, as in the invisible escaping waste heat of our furnaces. If, therefore, by the abatement of smoke development economic interests are not directly affected to the extent frequently supposed—the question generally being one of public interest—nevertheless both questions, the abatement of smoke and extreme utilization of fuels, are so intimately connected that, to obtain a wholesome progress, it will not be well to treat one without the other.

Smokeless working is, in general, more difficult the more we attempt to obtain economical combustion, hence, the more we strive to restrict the excess of air. These very conditions, therefore, constitute the *pith of the whole smoke question*, and in them lies the difficulty of the solution of the smoke question, if this is to be obtained in a simple manner, guaranteeing certainty in management and at the same time good utilization of fuel.

It is not sufficient to furnish a supply of air in keeping with the demand, but this supply must be so regulated that the minimum amount of air may be thoroughly mixed with the

gases before they have had time to cool down too much. We must not, for the sake of economy, demand the maximum decrease in smoke formation, but should rather be contented with a condition to be designated as *light smoke.*

NOTE REFERRED TO IN TABLE V.

These heat balances are based upon the heat values as found by the recalculation from the percentage of ash determined from average samples taken during the tests. The distribution of heat obtained from the original values is shown in Table VII.

The reason for making a recalculation was due to the large variations in the utilization and residue loss, as seen in Table VII—obtained from the duplicate tests I and II, as well as from those of III and IV—in spite of the slight differences in the factors of evaporation, and in the number of heat units utilized per kilogram, based upon the heat values of the samples.

There is undoubtedly ground for objecting to the recalculation, inasmuch as a complete and certain determination of the percentage of pure ash may be impracticable, a part of the ash being carried along in the flues. For fuels producing much, and specifically light ash, such as lignite, the amount of ash carried along in the flues will be considerable and hence would prohibit such a recalculation. However, in the present case, where the boilers were operated partly under a very light draft, the error in the determination of the percentage of ash could not have been large. Furthermore, the error would have to manifest itself similarly for all tests; consequently, *as a comparison for the residue losses*, which is the point in question here, the error would disappear. Very careful attention was, of course, given that the process of cleaning fires was effected at exactly the same time before beginning and before terminating each test. There was also no fluctuation of the load on the boilers during the various tests.

Therefore, the reason for the recalculation in the present case, where about 10,000 kilograms of coal were consumed dur-

Table VII.

Heat balance.	Trial 1.		Trial 2.		Average.		Trial 3.		Trial 4.		Average.		Trial 5.	
	T. U.	Per ct.	T. U.	Per ct.	T. U.	Per ct.	T. U.	Per ct.	T. U.	Per ct.	T. U.	Per ct.	T. U.	Per ct.
Utilized for making steam	**4,556**	67.5	**4,547**	64.8	4,552	66.1	**5,006**	71.9	**4,923**	75.2	4,965	73.5	4,789	72.4
Lost: (a) in the refuse	152	2.2	107	1.5	129	1.85	168	2.4	144	2.2	156	2.3	109	1.6
(b) due to heat carried away in the chimney	1,681	24.9	1,773	25.2	1,727	25 05	1,051	15.1	1,005	15.4	1,028	15 3	1,384	20.9
(c) other losses (residue) due to conduction, radiation, soot, unconsumed gases, etc	364	5.4	594	8.5	479	7	737	10.6	471	7.2	604	8.9	337	5.1
Total heat value of the fuel	6,753	...	7,021	...	6,887	...	6,962	...	6,543	..	6,753	...	6,619	...

Table VIII.

Number of trial	I		II.	III		IV.	Average of I to IV.	Variation per cent.	V	
Date of trial	March 4, 1903.		March 5, 1903.	March 6, 1903.		March 7, 1903.			June 18, 1903.	
.....	Sample 1.	Sample 2.	...	Sample 1.	Sample 2.	...			Sample 1.	Sample 2.
Carbon, per cent	70.2	72.18	73.9	74.2	72.63	68.4	...	...	70.43	70.04
Hydrogen, per cent	5.6	4.66	5.4	5.1	4.82	4.1	...	...	4.7	4.84
Oxygen and nitrogen, per cent	...	8.54	...	...	8.38	...	...	...	8.79	9.48
Sulphur, per cent	...	1.68	...	...	1.68	...	...	...	1.35	0.95
Ash, per cent	**9**	**5.88**	**4.6**	**5.3**	**6.16**	**10.5**	**6.99**	...	6.23	6.28
Moisture, per cent	7.5	7.06	7.1	7.3	6.33	7.1	...	...	8.5	8.41
Volatile matter, per cent	30.8	30.32	31.5	30.5	30.62	28.6	...	...	30.42	30.44
Fixed carbon, per cent	52.7	56.74	56.8	56.9	56.89	53.8	...	...	54.85	54.87
Heat value of the coal, T. U	6,643	6,862	7,021	6,987	6,936	6,534	6,830	+2.8 / −4.3	6,693	6,658
Heat value of combustible matter, T. U	8,000	7,930	**8,000**	8,044	7,969	**7,993**	**7,989**	+0.69 / −0.74	7,909	7,864
	7,965			**8,006**					7,886	
Ashes in total coal consumed, per cent	**6.96**		**7.6**	**6.1**		**6.05**	**6.68**	...	6.9	
Heat value on the basis of ash in total coal consumed, T.U.	6,787		6,781	6,944		6,899	6,853	+1.33 / −1.05	6,625	

Samples 1 and 2 from the trials of March 4 and 6, 1903, were analyzed in two different laboratories. The samples taken from the trial of June 18, 1903, were both analyzed in the same laboratory.

ing each test, may be seen by a comparison of the various columns. Table VIII contains the analyses and heat values of the average samples of the coal, as well as the calculated heat value of the combustible matter. In this table are the percentages of ash found from the total coal fired on each day of the tests, as well as the heat values recalculated on the basis of the percentage of ash, as used in Table V. The percentage of ash found in the samples varied, as is evident from the table, between 4.6 and 10.5 per cent., while the tests themselves gave a minimum value of 6.1 per cent., the maximum value being 7.6 per cent. From two samples, taken on March 4, 1903, and analyzed in two different laboratories, the percentage of ash amounted to 9 and 5.88 per cent., respectively, while 6.96 per cent. was obtained from the test on the same day. Considering the amount of coal consumed, the latter value is probably the more accurate.

That the heat values of the first four tests could not vary from each other at the rate shown in Table VII, is proved by the agreement of the heat values of the combustible matter as well as by that of the parts of the refuse by weight (Table V). Besides, the close agreement of the values for the heat utilized in T.U. could—especially in the case of tests I and II—hardly be possible if the actual heat values so largely disagreed, as would be the case according to the average samples.

Even if the reason for recalculating the series of tests given in Table I (for which Table III contains the tabulation of the coal samples) is not so apparent, nevertheless, it seems to me quite opportune to refer to these conditions at this point; for, from what has been said, it follows that the determination of the heat value in the usual manner, viz: by the sample test, even if very carefully carried out, still forms a considerable source of error, which may be the cause of many discrepancies; and it is, perhaps, not unjustifiable to agitate this matter, in order to find out if a greater degree of certainty could be obtained.

The heat value of the combustible matter in coal is, without doubt, obtained with sufficient accuracy by the usual method

in vogue. Therefore it would only be a question of determining the percentages of ash and moisture—which, of course, can vary considerably—with a greater degree of certainty, perhaps by using larger quantities for these examinations.

For considerable variations in the percentages of ash of the usual average samples, when compared to those values determined from the whole test, it seems to me—for comparative purposes—more accurate to base the comparison on the heat values as recalculated in the preceding manner, especially when sufficient care is taken in the determination of these values.

II. WAYS AND MEANS FOR THE ABATEMENT OF SMOKE DEVELOPMENT.

I now come to the second part of my paper, viz: the ways and means for the abatement of smoke.

Use of Special Fuels.

The simplest remedy for preventing the appearance of smoke is to use fuels that contain little or no volatile ingredients. In doing so we are not dependent upon the skill and attention of a fireman, and the ordinary grate, correctly proportioned, will be quite adequate.

Disregarding the fact that such fuels are not to be obtained in superabundant quantities, and therefore their extensive or compulsory use would, in general, greatly increase the cost, their use would in other respects be subject to certain restrictions. It is evident that fuel is less inclined to burn the fewer the volatile and easily ignited substances which are separated during combustion, therefore the less tendency there is to produce flame.

In cases where the variation in steam consumption is very marked and rapid, such fuels may fail, inasmuch as the heat development cannot follow with sufficient rapidity the increase in steam consumption, and therefore a constant pressure cannot be maintained, or for a decrease of the steam consumption the heat development cannot be checked rapidly enough.

Both of these reasons would prevent the use of such fuels for many installations. Anthracite and coke, for example, burn without forming flame, and can only be used for very uniform work, or require, if the steam consumption is somewhat variable, sufficiently large water volumes, *i. e.*, amply large, and therefore moderately worked boilers. On account of the slow rate of combustion, such fuels, of course, require more grate area for the same boiler performance, which likewise leads to boilers that are large, expensive and occupy much space. Such fuels also require a stronger draft.

It is not, however, intended to argue that where these fuels can be employed or where the conditions for obtaining them are favorable they should not be used. In view of the fact that special fuels make it possible to obtain smokeless combustion, or at least combustion accompanied by light smoke, entirely independent of the attendance, they come into consideration principally for furnaces where the prevention of smoke *must* be assured; for example, installations situated in localities in which there is much traffic, or adjacent to public squares, etc., where intense smoke development would be found especially annoying and burdensome. It is impossible, however, for industry to restrict itself to the exclusive use of such fuels; for, without regard to the cost, coals rich in gas are generally, from an engineering and business point of view, far more valuable than those poor in gas.

There is yet to be observed that in many cases mixtures of different kinds of fuels, for example, anthracite screenings and coal, have proven advantageous. Experience with this kind of fuel for water-tube boilers has been very satisfactory.

The Employment of Special Methods of Attendance and Special Furnace Arrangements.

For coal rich in gas the question is—as fully discussed in part one—to regulate, as much as possible, the supply of air for the varying needs during stoking, if combustion is to be effected with slight smoke development and at the same time economically.

As long as the coal consumption is moderate this can be attained in a satisfactory manner with the ordinary grate by careful working; the fuel being thrown only on the front part of the grate, after having pushed back the glowing coals. In this way the gas is slowly abstracted from the coal; the air required will vary but little, and it will not be difficult to obtain combustion accompanied by light smoke for a relatively slight excess of air. Smoke development can also be restricted by a uniform spreading of the coal in small quantities. But experience has shown that a larger excess of air will be necessary as the variation in the amount required is greater, and therefore the supply of air, if sufficient during the abstraction of gas, will subsequently be too large.

The moderate coal consumption required for the success of these methods of working naturally presupposes large installations, the existence of which no doubt would, in general, be in the interest of extreme utilization of fuel.

The greater the coal consumption the more difficult it becomes to work the ordinary grate with a moderate supply of air without developing considerable smoke, since more coal must be added in the various charges, or, if thrown near the front of the grate, the coal must be pushed back before gas abstraction has been completed. Therefore neither of these operations could be performed, without the production of smoke, should the air supply suddenly become insufficient.

The rate of combustion, especially when using coal rich in gas, must not fall below a certain limit, for without an excess of air the draft will be too light, the result being a smouldering fire and imperfect combustion.

As is well known, to assist the fireman in bringing about perfect combustion and at the same time to be less dependent on his skill, a great many contrivances have been invented. They can, in general, be divided into two main groups. One group endeavors to regulate the supply of air uniformly, either by continuous stoking or by causing first a slow abstraction of the gas from the coal, during which the volatile substances flow off over the glowing coals or through them. To this

group belong furnaces with mechanical stokers, coal-dust furnaces, furnaces with inclined grates, basket-grate furnaces and similar arrangements. The other group retains the continuous stoking feature, and seeks to meet a possible variation in the air required by regulating the supply by the introduction of so-called secondary air in the burning gas stream immediately after the stoking and working of the fires.

I am well aware that many manufacturers are very suspicious of any special contrivances for restricting smoke development, and, as a matter of fact, this is fully justified in the case of a great many of those that have been proposed.

Apart from the failures caused by such faulty constructions, aversion to these contrivances has been frequently produced through the exorbitant claims for economy made by the over-zealous inventors. With respect to durability and safety in operating, much is yet desired. Frequently proper consideration is not given to the fact that the effectiveness of any arrangement is limited to certain conditions of manufacture; and, finally, disappointment has been caused by the fact that, even in well constructed furnaces, success is very much dependent upon the diligence of the fireman.

It is not my object to discuss fully these various constructions, though I would like to give the characteristics of the main groups mentioned.

For the large class of flue boilers having internal furnaces, for example, arrangements with upper-draft air supply occupy a prominent place and are not to be underrated. They are based on the well-known principle of providing the large quantity of air required for combustion immediately after stoking or working of the fires, by simultaneously introducing it, above and at the back of the grate, directly into the burning gas stream. For this, of course, the greatest variety in arrangement is possible. The air can be introduced at the front through the fire door or through special openings; it can be introduced at the sides of the grate, or the bridge wall can be used for this purpose (in which case it must be of a

perforated construction); or, finally, it can be introduced in a special combustion chamber, built back of the bridge wall.

Even if these arrangements, without regard to their durability or adaptability, are not all equally effective (but will require a more or less large excess of air, according to the manner in which it is supplied), nevertheless the widely-accepted opinion that they *all* tend to produce light smoke does not quite agree, in the majority of cases, with the facts.

The effect of the upper-air supply can be easily demonstrated by leaving the fire door slightly open immediately after working the fires and as long as the main gas generation continues. I have caused this test to be frequently made during consecutive analyses of the waste gases and, when properly carried out, no impairment of the utilization was apparent, while the abatement of smoke was quite manifest.

Tables X and XI.

Data taken at the end of boiler.	Carbonic acid.	Oxygen.	Temperature of the gases.	Temperature of the air.	Draft, water column.	Waste heat loss.
Table X.	*per ct.*	*per ct.*	°C.	°C.	*mm.*	*per ct.*
Examination	8.8	10.	289	17	14.8	20.2
In charge of fireman instructor	12.8	6.1	292	16	9.5	14.
regular fireman after being instructed	11.7	7.3	276	14	8.4	14.5
First inspection	13.1	5.7	310	23	9.8	14.3
Second inspection	13.8	5.	314	31	8.	13.3
Table XI.						
Examination	8.	.....	339	20	11.	26.2
In charge of fireman instructor	11.	.....	326	20	4.	18.4
regular fireman after being instructed	11.4	.....	317	20	3.5	16.9
First inspection	10.9	8.4	335	28	3.5	16.2
Second inspection	12.4	6.6	349	13	6.	18.
Third inspection (lighter duty)	12.	7.3	239	18	2.5	12.

Relative to this, there is given in Table X an example of waste gas analysis, which clearly shows that, compared with the pernicious influence of the existing inadequate method of working (discussed in the foregoing), there can be no question of impairment of the utilization by supplying air through the fire door or in any other way, provided it is done effectively

and at the proper time. For the conditions as they existed in the case which forms the basis of Table X, the average percentage of carbonic acid found at the end of the boiler was only 8.8 per cent., and the waste heat loss was about 20 per cent. At the same time the coal, which was of a quality strongly inclined to gas producing, gave off dense smoke after stoking and working of the fires. By expert working and regulating the supply of air in the manner indicated, an average proportion of carbonic acid of 12.8 per cent. and a decrease in the loss of the waste heat of about 14 per cent. could be obtained, the smoke making being quite moderate. From this it follows that it was not a question of smoke abatement alone, but that perfect combustion also set in for a moderate supply of air. After this a fireman-instructor was put in charge. Similar results, however, were also obtained from a test made with the regular firemen—after having been instructed, but entirely uninfluenced while in charge—as well as during subsequent inspections.

This remedy—leaving the furnace door slightly open to supply the necessary air—can, of course, be regarded only as a makeshift, for it demands of the fireman, especially if he has to look after several furnaces, very much attention. If the door remains open longer than necessary, an excess of air results and, consequently, an increase of the waste heat loss, as in the case of improper stoking.*

To obviate this a great many automatic devices have been invented, which at certain intervals of time admit the necessary air by means of a gradually-closing damper, correctly timed.

As a matter of fact, very satisfactory results can be obtained with such devices if correctly adjusted and expertly managed, as was demonstrated in a series of tests carried out by our association.†

* For this the fireman must be in a position to observe the top of the smokestack, an arrangement which, it may be said, should always be made.

† This association, for the purpose of clearing up certain questions (among which was that of the value of the upper draft), has carried out thorough tests under the most varying conditions. An installation was especially provided, for which an allowance of 6,000 marks was granted on the occasion of the anniversary of the German industry, and a report will be shortly forthcoming.

In regard to the question of utilization in these furnaces, as compared with the ordinary grate, without doubt some inventors go too far. It is generally overlooked that even if on the one hand more perfect combustion is obtained, on the other hand the air supply, though the method of working remain the same, is somewhat increased, and with this the waste heat loss is greater; it follows from this that utilization can increase only when the former influences predominate.

Therefore it will not generally be a question of any *very* great differences; but, if actual differences have been established, they are generally not to be attributed to the principle of the furnace, but to a different method of working adopted by the fireman or to a more sparing supply of air (provided, of course, everything else has been correctly carried out as de-described in the foregoing).*

Furthermore, even if such devices make it easier for the fireman to work, with less smoke development than with the ordinary grate, the fact is self-evident that their effectiveness presupposes expert management. Their advantage, however, is in making it possible when using coal rich in gas to operate with a moderate excess of air, without incurring the danger of dense smoke making, which would be unavoidable in the case of failure of the secondary air supply under similar conditions. From an economical point of view such arrangements are not less dependent upon the fireman than the ordinary grate.

As soon as it becomes necessary to work with a greater supply of air to obtain perfect combustion more heat escapes—invisible, of course, but in any case not useful—through the stack, and the utilization suffers. After all, it cannot be maintained that such furnaces, without expert attendance, are less economical than the ordinary grate under the same conditions. In the numerous tests carried out by our Association we have, of course, encountered many such installations operating with bad utilization, where the average proportion of carbonic acid at the end of the boiler was 6 and 7 per cent., but still not more frequently than in the ordinary furnace

*See especially p. 656, last paragraph and the following.

grate operated in the same manner. On the whole, it was shown that an average proportion of carbonic acid of 10 to 12 per cent. at the end of the boiler could be easily obtained in connection with satisfactory conditions of smoke making.

A further advantage of this type of furnace is the simple manner in which it can be combined with the ordinary grate and the good qualities of the latter be retained. They also permit a reasonably high coal consumption without losing any of their smoke-preventing qualities.

The main disadvantage of furnaces with an upper-air supply is that by incorrectly adjusting this draft more air is admitted than necessary, thus causing a loss in economy. This can be met by regular and expert supervision, as practiced by the Association for the Management of Furnaces and the Controlling of Smoke, of Hamburg.

As an example, I have chosen the results of an installation (to which might be added a series of similar ones) given in Table XI. The problem given here was that of four flue boilers equipped with Kowitzke furnaces. In connection with this it should be stated that the inspections were made unexpectedly, and the examination of the installation was made in the condition found at the time. The values clearly show that it was possible to work with quite a moderate excess of air and with satisfactory utilization. The combustion was satisfactory, producing a light smoke.

Any defects resulting from a change in the working force, different kinds of coal, or varying conditions of management, can be easily removed in this manner by frequent inspections, which the association undertakes to make once in every four months.

For the other large groups of furnaces having either continuous or interrupted stoking the process of combustion is not disturbed. In this case the excess of air naturally can be more readily diminished, without questioning the attainment of perfect combustion, so that generally from these arrangements higher values of utilization are to be expected than

from the ordinary grate or from constructions that are only additions to the latter.

As a matter of fact, among these are to be found a series of thoroughly tried furnaces. I need only recall without entering into details the various arrangements of the Tenbrink furnace, inclined-grate furnaces, and similar constructions. In most of these furnaces is has been shown, especially for fuels strongly inclined to gas producing, that an upper supply of air into the burning gas stream is practicable. For the uniform combustion to be had in these furnaces such supply can be easily regulated.

The same applies to furnaces having mechanical stokers. Unfortunately, the useful constructions of this class—requiring infrequent repairs, and thus maintaining their good qualities—can, as a rule, be used only for coal of a uniform quality. If they could be made to burn, successfully, the ordinary run-of-mine coal, the question of their use would be of considerable importance, especially for large installations, where a smaller working force would be required.

These constructions are, in themselves, of course, more expensive than the ordinary grate, besides the additional attachments required. They also call for boilers partly of special construction, or require an increased expenditure for maintenance. The greater cost, it is true, is generally counterbalanced by a better utilization; but the paying qualities will always depend materially on the cost of the coal, and for this reason these arrangements have been largely introduced into localities where coal is expensive.

That a more perfect arrangement also requires to be properly handled and supervised, if it is to work continuously and in a reliable manner, is certainly a well-known fact, and this is of such importance that we cannot afford to neglect it.

It would be conducive to a wholesome progress if inventors would, first of all, relinquish the idea of improving the economy while trying to prevent smoke. At any rate more account should be taken of the actual conditions, which, I believe, I have to a certain extent set forth, and which indicate that the

main losses are not to be sought in the smoke; consequently it should be considered satisfactory if we are able to bring about combustion accompanied by light smoke *at the same time preserving the economy.* If the economy is raised, which in many cases is not impossible, so much the better.

As long as coal is burned we shall never obtain a universal furnace that will meet the varied requirements, owing to the many properties of the fuels, to the peculiarities inherent in management, to different types of boilers, and to local conditions. In like manner, we need not anticipate new constructions obviating the making of smoke in a greater degree than heretofore.

Moreover, it must be particularly pointed out that the effectiveness of all furnace arrangements for smoke abatement are dependent upon the injurious effect on the boiler, and that even the best arrangements can fail if the rate of combustion (*i. e.*, the coal burnt per square meter of grate surface in a unit of time) exceeds certain limits.

Still, good results can yet be obtained, for certain conditions, with various arrangements; but to exceed these conditions, even in the interest of economy, does not recommend itself.*

The question of attendance is of vital importance for all

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* Dense smoke formation is very often mainly attributed to overloading of the boiler. In reality there are not many installations in which this condition is such that it would not be possible to bring about satisfactory conditions through suitably designed furnaces and *expert* attendance. For example, coal containing much gas can be burnt on an ordinary grate, in a flue boiler, at the rate of 150 kg. per sq. m. of grate surface per hour with a moderate excess of air, and little smoke, provided there is arrangement, properly regulated and handled, for an upper-air supply.

Increasing smoke formation with an increase of the load in a boiler installation can in most cases be traced to the fireman who keeps the grate well covered to satisfy the increased demand in steam consumption; in so doing he works with a smaller excess of air than would be the case if the load were lighter. If care is not taken after stoking to provide a proper supply of air, then dense smoke formation cannot be avoided.

Similar conditions obtain in the case of the draft. Installations in which the draft is so meager that the quantity of air required for perfect combustion could not be admitted are rarely found, while the condition of working with too much draft is nearly always the rule.

The increase in smoke formation with increasing load can often be traced to a decrease in the excess of air; therefore the bringing about of perfect combustion without proper expedients is impossible.

The case is by no means rare where, despite an increase in smoke formation, utilization improves with an increasing load by reason of a reduced excess in air. Such improvement occurs also when working, as is frequently the case, with a light load and full draft of the stack, *i. e.*, with the damper wide open.

furnace arrangements. Upon this question I have dwelt at length for furnaces having a secondary air supply.

Education and Supervision of the Firemen.

Correct training and, more particularly expert supervision and control of the fireman, is of fundamental importance in the whole question of management of furnaces.

The Association for the Management of Furnaces and Controlling of Smoke, of Hamburg, takes hold of this question in the following manner:

"After all the conditions have been thoroughly examined by an engineer, a fireman instructor, through his own efforts and example, shows the fireman on the ground what may be obtained for the special conditions. Based upon the results thus obtained, the firemen are then placed under the direction of a fireman instructor until they have thoroughly mastered the various manipulations. After this the examination is repeated by the engineer, and the results noted, the progress made being shown the firemen, and their knowledge of the correct management of a furnace thus increased. At the same time, the association seeks to impress upon the managers or others in charge the need of supervision and systematic control to prevent relapse into former bad practice.

"In this, we believe, we are on the right path and, in any event, attain more in this manner than through fireman schools, fireman examinations," etc.

The establishment of schools for firemen was also considered by our association in Hamburg; but I am of the opinion that the effect of such schools upon the management of furnaces will not be as marked as is generally expected. The whole question is not so much one of knowledge as of a certain, easily-obtainable, skill; *and more particularly is it a matter of reliability and constant attention to the main points at issue.*

Therefore, owing to the shortcomings of mankind, without a constant and expert supervision, permanent success will never

be achieved. Such supervision would continually tend toward, and produce rules for, correct working, and it would also take due cognizance of any variation in the conditions.

Similar instructions repeatedly given in specially selected installations are of course much more simple and less tedious, full regard being had to the peculiarities of management in each case, than instructing individually on the ground. However, it is an ever-recurring experience, with the varying conditions exhibited in this field, that firemen, in most cases, can only be convinced by showing them in their own installations with its individual peculiarities, that improved working is possible.

It is likely that courses of instruction in this branch would meet with more success if they were given to those in charge of firemen. For in these very circles it is of the utmost importance that the understanding of the procedure for the management of furnaces be better than it is at present, as success depends very materially on the expert, and, if necessary, determined manner of supervision during working. Influence in this direction is also to be obtained in a thoroughly satisfactory way along with the method adopted by our association for the education of firemen.

In regard to the duty of fireman instructors, I wish to say further that fireman instructors have now been in vogue elsewhere for some length of time. But my observation of this institution has convinced me that if anything really substantial is to be done, the efforts of the fireman instructor must be guided by a trained engineer. As a rule, the fireman instructor himself can only be one who carries out orders. In the case of the judgment of furnaces—simpler installations excepted—problems are presented which demand for their solution more knowledge and ability than can be expected of a fireman instructor who comes from the working classes.

I, therefore, proceeded in such a manner as to have the association engage, first, one scientifically-trained engineer and later on another; both were familiar with the management of boilers, and besides, they were required, for some

time, to work as firemen in various installations. Each of these engineers is given a fireman instructor, and the necessary examinations, instructions, etc., are carried out by them together. I believe the results thus far obtained have justified this method.

III. ECONOMY GAINED IN SUPERVISING THE MANAGEMENT OF FURNACES.

It is my wish to mention the importance—not to be underrated—of the examination and supervision of installations when carried out in the manner indicated; and I would also especially refer, from the view point of economy, to the analysis of the waste gases.

From the first part of my subject it is clear that the observation of the top of the smoke stack alone in no way suffices to form an adequate judgment of the working of a furnace, but is of real value only when combined with a correct analysis of the waste gases.

Of course these analyses are little comprehended at first. It is recalled that the factor of evaporation is something more tangible, and that the knowledge of the same for judging the conditions in a practical way is sufficient. But this is only correct up to a certain point; for, apart from the fact that reliable determination of the factor of evaporation is comparatively circumstantial, and from the further fact that many sources of errors existing in establishments are unknown to us, its value is always to be regarded only as a relative one. Its importance is by no means to be misjudged, but taken alone, without simultaneous analysis of the waste gases, it never shows whether the fuel in an installation has been actually utilized, as might be reasonably expected; while this always follows from waste gas analyses. These analyses not only show, in a simple manner, about how much is yet to be gained under the conditions existing, but with their assistance it is also comparatively easy to establish whether the cause of inefficient utilization results from the method of working adopted by the fireman, badly proportioned grate surfaces, porous brick-

work, poorly cleaned heating surfaces, errors in the lead of the gases, or the conditions of the draft, etc.

As a matter of fact, the waste gas analysis when correctly carried out is an *extremely valuable means of supervision* for actual practice. They hold at least the same relation to the furnace as an indicator test for a steam engine.

As to the significance, however, of the waste heat loss for the utilization, which can only be found in this manner, we are, in general, not yet well informed.

The opinion often prevails, even in well arranged establishments, that so long as there is no smoke made, everything else is satisfactory. In this we either forget, or else we are not clear upon the subject, that a much greater excess of air might exist (even in the case of a good arrangement) than would be necessary to obtain, when working with efficient attendance, light smoke. Hence, a regular supervision of the management of furnaces by means of waste gas analysis is in any case very valuable, for only thereby can it be established whether the fire in relation to the smoke and in relation to the utilization of the fuel is properly managed.

IV. DOMESTIC FURNACES.

As already mentioned in the introduction, in many cities, during the winter months, responsibility for the smoke nuisance rests as much with the domestic furnace as with furnaces of industrial establishments.

A partial relief can of course be obtained in this field by the use of fuels making light smoke, such, for example, as lignite briquettes, which are much used in Berlin.

Coke and anthracite are also well adapted for domestic purposes. But, despite the economical arrangements to be had for the latter fuels in the self-feeding stoves, their use, as well as those of briquettes, is already limited, owing to the fact that in many places they are considerably more expensive than coal and an increased demand would raise their cost. It would also be impossible to obtain them everywhere in the requisite quantities for general use.

By improving the construction of the stoves something, to be sure, might yet be obtained; but, precisely as in the case of the large furnaces, the effectiveness of these small stoves, even for the best constructions, depends on expert attendance. This is, however, for the class of employés in question, not to be expected in an adequate degree, and a successful influence in this direction would be impossible.

Without doubt more can be obtained from central heating stations, which are constantly becoming more numerous, despite the fact that detached furnaces burning coal are inefficient as well as inconvenient. In the smaller of these central stations coke furnaces are generally used on account of the low percentage of gas contained in this fuel and the attendant simple and safe working conditions, with no smoke nuisance. In the case of the larger of these central stations—the heat generators being generally steam boilers, with large furnaces requiring constant attendance—they can be treated in the same manner as the furnaces of regular industrial plants.

Perhaps the present competition between the gas engine and steam engine may result in producing a cheaper gas for general use. This would make a more extended use of gas possible for cooking and heating, forming, of course, the most effectual relief.

To briefly recapitulate the ways and means for the abatement of smoke in industrial furnaces we have the following points:

1. *Education and supervision of the firemen.*

2. *Improving the furnaces with respect to construction and management.*

3. In case nothing can be attained in any other manner, or if we want to be absolutely certain, then *the use of fuels producing light smoke.*

None of these ways, taken alone, is wholly satisfactory, but by taking them all together, and perseveringly pursuing them, without doubt progress can be made.

At this point, gentlemen, I have, practically, concluded my address, but I wish to add a few general comments.

My deductions may have shown that, as respects smoke abatement, nothing satisfactory is to be obtained by way of a smoke prohibition law, because the conditions for that are entirely too complicated. If, furthermore, industrial interests should not only object to the compulsory use of certain kinds of fuels, but also to a general law for the employment of special furnace arrangements, then, in my opinion, owing to the conditions which prevail in this field, such objections would be, in a great measure, justifiable. The question is not only one of providing such special arrangements, *but they must also be adapted to the existing conditions, and, further, must be correctly managed and intelligently supervised.* The latter point—skilled and intelligent supervision—is of the greatest importance for the entire success of these arrangements. Therefore, the authorities would also have to adopt the needful precautionary measures in this direction if such laws are to be of any benefit. But considering such municipal measures, and what has been, and might yet be accomplished by them, I believe that an easier and surer solution would be obtained through concerted and harmonious action on the part of the manufacturers interested; that is, a policy of mutual help, such as pursued by the Hamburg association. The management of these combined interests should be assured of an opportunity to exert educational pressure, and also receive the confidence and support of the city authorities, such as is given to the societies for boiler inspection. All this would be most conducive to the aims of such a union among the manufacturers; but it is evident that, besides, the management should possess the power to act in an advisory capacity in the case of any new installations made in establishments which come under its supervision.

For industry, the way here outlined has the absolute advantage, that (except in some cases, where the use of special arrangements cannot possibly be avoided) full attention is given to obtain extreme utilization of the fuels, and the owner is

guaranteed that all conditions are duly considered. Moreover, the owner may command at any time expert advice on all questions relating to the management of furnaces; his plant in regard to economy will be regularly controlled, and his employés will be constantly guided and instructed. But such matters as these (and they are not matters for the municipal authorities either to consider or settle) could all be readily solved by the officers of the association, owing to their close and constant contact with the various establishments and knowledge of peculiarities of management, etc. It would not be difficult for them to discover the means needed to restrict smoke development, without exposing the owner to the risk of costly experiments.

With this I shall conclude my remarks. Owing to the vital nature of this problem, the industrial world is much interested in the proper method of procedure, and in my discussion I have endeavored to describe the start which our association has made, and give an idea of the work already accomplished along the lines pursued. My statements will have achieved their end if, together with the association's record up to the present, some profit to local interests results, and an incentive is given elsewhere to further progress in dealing with this important question for the general welfare.

THE U. S. S. *GALVESTON.*

GENERAL DESCRIPTION—STANDARDIZATION, OFFICIAL AND ENDURANCE TRIALS.

The *Galveston* (*Cruiser No. 17*) is one of six sheathed, protected cruisers, known as the *Denver* type, authorized by Congress in 1899.

The contract for the *Galveston* was awarded to the William R. Trigg Co., of Richmond, Va., December 14, 1899, who were to deliver her completed in thirty months, or in June, 1902. Contract price, $1,027,000. Owing to various delays, due to non-delivery of material, the time for completion was extended to September 15, 1903. On December 24, 1902, the W. R. Trigg Co. failed, and the contract was declared forfeited May 14, 1903. A board of naval officers was appointed to make a complete inventory of all work done or commenced and of all material on hand, for the purpose of completing the vessel.

The Board found the hull 66 per cent. and the machinery 77.25 per cent. completed.

In September, 1903, the vessel was removed to the Navy Yard, Norfolk, Va., where she was completed by the Government, and commissioned February 15, 1905.

As complete descriptions of this class of ships have appeared at various times in the JOURNAL, it will be only necessary to repeat here the principal dimensions. For more extended data, the reader is referred to Vol. XV, page 1110 (Trial of U. S. S. *Cleveland*), Vol. XVI, page 1 (U. S. S. *Tacoma*), page 67 (U. S. S. *Denver*, page 227 (U. S. S. *Des Moines*), and Vol. XVII, page 104 (U. S. S. *Chattanooga*).

HULL.

The hull is constructed of mild steel, the outer plating being composed generally of 15-pound plating. There are one hundred transverse frames, spaced 3 feet apart, and formed of

Z-bars (frames 18 to 36), 6 inches by 3½ inches by 3½ inches by 15½ pounds, except forward and aft (frames 1 to 17 and 87 to 100), which are 4 inches by 3½ inches by 9.1 pounds angle bars. The outside plating is sheathed with Georgia pine, 4 inches thick, extending to about 42 inches above the water-line amidships, shearing up forward to 69 inches, and aft to 51 inches.

Length L. W., feet	292
over all, feet and inches	308- 9
Beam molded, feet and inches	43- 3¼
extreme, feet	44
Mean draught (with 467 tons coal and two-thirds stores), feet and inches	15- 9
Corresponding displacement, tons	3,200
Area of midship section, square feet	615.2
load-water plane, square feet	9,370
Tons per inch of immersion	22.3

ARMAMENT.

Main battery	10 5-inch R. F. guns, 8 6-pdrs.
Secondary battery	2 1-pdrs., 4 Colt's automatic.
Mean height of axis of 5-inch guns:	
Main deck, above load-water line, feet and inches	21- 5
Gun deck, above load-water line, feet and inches	12- 4

MACHINERY.

There are two triple-expansion, four-cylinder engines, of the vertical, inverted type, placed abreast of each other in separate watertight compartments, and designed to develop a total of 4,500 I.H.P. at about 172 revolutions. The engines are right-and-left, turn inboard, and have the following sequence of cylinders, beginning forward: Forward low pressure, high pressure, intermediate pressure and after low pressure. There are two main condensers with a cooling surface of 3,004 square feet each.

Steam is supplied by six water-tube boilers of the Babcock & Wilcox *Alert* type.

ENGINES.

Number of engines	2
cylinders, each engine	4
Stroke (common), inches	30
Diameter H.P. cylinder, inches (originally 18 inches)	18½
I.P. cylinder, inches	29
L.P. (2) cylinders, inches	35½

Cylinder clearances :	*Stbd. engine.* Top.	Bottom.	*Port engine.* Top.	Bottom.
H.P. cylinder, inch	$\frac{7}{32}$	$\frac{7}{16}$	$\frac{5}{32}$	$\frac{7}{16}$
I.P. cylinder, inch	$\frac{9}{32}$	$\frac{7}{16}$	$\frac{9}{32}$	$\frac{13}{64}$
Forward L.P. cylinder, inch	$\frac{13}{32}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{1}{2}$
Aft L.P. cylinder, inch	$\frac{31}{64}$	$\frac{15}{32}$	$\frac{1}{2}$	$\frac{15}{32}$

Valves, H.P. (one piston for each cylinder), diameter, inches 11
I.P. (two for each cylinder), diameter, inches 11
L.P. (one double-ported slide for each cylinder), inches, length, 25¼; width, 39½; port, length 37

Table I.—DETAILS OF VALVE SETTINGS.

	H.P.		I.P.		L.P.	
Side on which steam is taken	Inside.		Outside.		Outside.	
Travel of valve, inches	4		4		4	
	Top.	*Bottom.*	*Top.*	*Bottom.*	*Top.*	*Bottom.*
Width of port, inches	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}\times37$	$1\frac{1}{4}\times37$
Steam lap, inch	$\frac{29}{32}$	$\frac{27}{32}$	$\frac{29}{32}$	$\frac{27}{32}$	$\frac{1}{16}$	$\frac{1}{16}$
Exhaust lap, inch	$-\frac{1}{16}$	$-\frac{1}{16}$	$-\frac{1}{16}$	$-\frac{1}{16}$	...	$+\frac{1}{4}$
Angular advance, degrees	35½	35½	35½	35½	40	40
Steam lead, angular, degrees	8	10	8	10	12	16
linear, inch	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{11}{32}$
Cut-off, in inches	$23\frac{3}{8}$	$21\frac{3}{16}$	$23\frac{3}{8}$	$21\frac{3}{16}$	$22\frac{1}{16}$	$20\frac{3}{16}$
decimal of stroke	0.78	0.705	0.78	0.705	0.735	0.672
Exhaust release, in inches	$3\frac{7}{16}$	$3\frac{3}{8}$	$3\frac{7}{16}$	$3\frac{3}{8}$	$2\frac{13}{16}$	$2\frac{7}{8}$
decimal of stroke	0.114	0.1205	0.114	0.1205	0.093	0.095
Compression, in inches	$1\frac{7}{8}$	$1\frac{15}{16}$	$1\frac{7}{8}$	$1\frac{15}{16}$	$4\frac{3}{16}$	$3\frac{1}{4}$
decimal of stroke	0.062	0.064	0.062	0.064	0.139	0.129
Steam opening, inches	$1\frac{3}{32}$	$1\frac{5}{32}$	$1\frac{3}{32}$	$1\frac{5}{32}$	$1\frac{1}{16}\times2$	$1\frac{3}{16}\times2$
Exhaust opening	Full port.		Full port.		Full port.	
Velocity of steam, in feet per second, (860 ft., 172 revs., per minute)	128.6	121.6	166.9	157.9	189.3	169.4
Velocity of exhaust, in feet per sec., (860 ft., 172 revs., per minute)	93.82	93.82	121.7	121.7	134.12	134.12
Velocity of exhaust to condenser	115 6 feet per second, 12½ nozzle.					

PROPELLERS.

Starboard	Right hand.
Port	Left hand.
Number of blades	3
Diameter, feet and inches	10- 6
Pitch, feet and inches (as set on trial)	11- 8½
adjustable, feet and inches	11- 2 to 12- 5
feet and inches (as designed)	11- 9
Helicoidal area, each screw, square feet	34
Projected area, each screw, square feet	29
Disc area, square feet	86.6
Pitch ratio, $\frac{P}{D}$	1 : 115

BOILERS.

Type	Babcock & Wilcox, marine.
Number of boilers	6

Length, feet and inches	9–10$\frac{1}{2}$
Width, feet and inches	9– 0$\frac{1}{2}$
Height to center of steam drum, feet and inches	10– 3
Diameter of steam drum, inside, inches	42
Length of steam drum, feet and inches	9– 6$\frac{5}{8}$
Thickness of steam drum, inch	$\frac{7}{16}$
Number of furnaces per boiler	1
furnace doors per boiler	3
Length of grates, feet and inches	6– 6
Width of grates, feet and inches	7– 7
Diameter of tubes, feet and inches	2 and 4
Thickness of 2-inch tubes	No. 8 B.W.G.
4-inch tubes	No. 6 B.W.G.
Number of 2-inch tubes per boiler, 8 feet 11 inches long	406
4-inch tubes per boiler, 8 feet 11 inches long	17
Number of circulating tubes per boiler, 7 feet 3 inches long	15
Total number of 2-inch tubes	2,436
4-inch tubes	192
Heating surface per boiler, square feet	2,200
Total heating surface, square feet	13,200
Grate surface per boiler, square feet	50
Total grate surface, square feet	300
Ratio heating surface to grate surface	44 : 1
Tube-heating surface per boiler, square feet	2,172
Total tube-heating surface, square feet	13,032
Plate-heating surface per boiler, drum, square feet	28
Total plate-heating surface, drums, square feet	168
Number side boxes, 8 feet 3 inches $\times$ 5$\frac{5}{8}$ $\times$ 5$\frac{5}{8}$ O.S. by $\frac{5}{8}$-inch thick per boiler	8
Number side boxes, 8 feet 3 inches $\times$ 5$\frac{5}{8}$ $\times$ 5$\frac{5}{8}$ O.S. by $\frac{5}{8}$-inch thick per boiler, total	48
Number corner boxes, front, per boiler, 38$\frac{7}{16}$-inch $\times$ 5$\frac{1}{4}$-inch I.S. $\times$ $\frac{5}{8}$-inch thick	2
Number corner boxes, front, total	12
Number corner boxes, rear, per boiler, 39$\frac{13}{16}$ inch $\times$ 5$\frac{1}{4}$ $\times$ 5$\frac{1}{4}$ inches I.S. $\times$ $\frac{5}{8}$-inch thick	2
Number corner boxes, rear, total	12
Number front cross boxes per boiler, 7 feet 6$\frac{1}{4}$ inches $\times$ 5$\frac{1}{4}$ inches $\times$ 5$\frac{1}{4}$ inches I.S. $\times$ $\frac{5}{8}$-inch thick	1
Number front cross boxes, total	6
Number of smoke pipes	2
Diameter of smoke pipes for two forward boilers, feet and inches	4– 4
Diameter of smoke pipes for four after boilers, feet	6
Total area of smoke pipes, square feet	43
Height above grate bars, feet	70
Ratio smoke-pipe area to grate surface	1 : 6.98
Pressure for which designed, pounds per square inch	275

Table II.—AUXILIARY MACHINERY U. S. S. *GALVESTON.*

Purpose for which used.	No.	Type.	Make.	Diameter steam cylinder.	Diameter water cylinder.	Stroke.	Weight of each.	Remarks.
				inches.	*inches.*	*in.*	*pounds.*	
Main air pump.....	2	Single-acting.	Trigg.	.....	20	10	.....	Worked by beam, forward L.P. crosshead.
Comb. aux. air and circulating pump...	2	Double-acting.	M. T. Davidson.	6	Air 7 / Water 10	10	1,981	For auxiliary condensers.
Auxiliary condenser..	2	.....	.....	.....	.....	...	2,402	400 sq. ft. cooling surface, each.
Main circulat'g pump	2	Centrifugal.	Trigg.	(2) 6	Runner 36	6	.....	Suction and discharge diameter, 12 inches.
Feed pumps	4	Vert. simplex.	Blake.	10	7	12	2,070	Outside-packed plunger.
Fire and bilge pumps.	4	Vert. simplex.	Blake.	7	7	12	1,015	2 in engine rooms, 2 in fire rooms.
Water-service pumps.	2	Vert. simplex.	Blake.	7	7	12	1,057	1 each engine room.
Hotwell pumps.....	2	Vert. simplex.	Blake.	7	7	12	1,045	1 each engine room.
Shaft bilge pump.....	2	Single-acting plunger.	Trigg.	.....	3	8	202	On forward end of shaft.
Evap. feed pump.....	1	Vert. simplex.	Blake.	4½	3½	6	.....	
Forced-draft blowers.	3	Fan.	B. F. Sturtevant.	(2) 5	Fan 60	4	2,467	
Dynamo and engine..	4	Compound.	W. D. Forbes.	7½-13	.....	7½	8,412	Direct connected to Bullock 25 kw., 80 volt generators.
Capstan and anchor engine.....	1	Double.	Hyde.	(2) 9	.....	9	18,900	
Steering engine.....	1	Double.	Williamson.	8	.....	10	13,005	
Distillers	3	.....	Bureau design.	.....	.....	...	556	2,500 gallons each.
Ash-hoist engine.....	2	.....	Williamson.	4½	.....	4½	.....	
Evaporators.....	2	.....	M. T. Davidson.	.....	.....	...	6,297	5,000 gallons each.
Ice machine.....	1	.....	Allen dense air.	7	.....	10	5,950	One ton capacity.

MACHINERY WEIGHTS IN DETAIL.

Actual weights of the various parts of the finished machinery, as built and installed, are as follows:

Group.	*Tons.*
I. Main engine cylinders, receiver pipes, etc	26.55
II. Shafting	20.15
III. Main engine framing and bearings	23.52
IV. Main engine reciprocating parts	7.23
V. Main engine valve gear	6.53
VI. Main condensers	12.91
VII. Main air and circulating pumps	5.76
VIII. Propellers	5.32
IX. Boilers	97.97
X. Boiler fittings	37.70
XI. Smoke pipes and uptakes	24.70
XII. Steam and exhaust pipes and valves	15.05
XIII. Suction and discharge pipes and valves	18.19
XIV. Lagging and clothing	8.15
XV. Flooring, gratings, etc	15.26
XVI. Auxiliaries	18.43
XVII. Fittings and gear	15.77
XVIII. Water in boilers and machinery	37.10
XIX. Stores, tools and spare parts	19.20
XX. Miscellaneous machinery, heating, distilling and refrigerating plants	28.76
XXII. Connections under steam engineering to other miscellaneous machinery	3.78
Total	448.03

STANDARDIZATION OF SCREWS.

Progressive runs were made over a measured mile for the purpose of standardizing the screws and thus determining the number of revolutions to be made by the engines in order to give the speed for four hours at sea, required by the contract. The course off Provincetown, Mass., was selected for this purpose, and the runs with and against the tide made at the speed shown on Table III, from which the curves on plates 1 and 2 were plotted.

Table IV gives the general performance data as recorded or calculated.

The runs over the measured mile were made in a smooth sea, with calms to light southerly airs during the first eight runs and a gentle southwest breeze during the remainder.

Table III.—U. S. S. *GALVESTON.*

Data from Progressive Runs over Measured-Mile Course off Princetown, Mass., May 29, 1905.

Number and direction of run.	Time on course.	Revolutions of main engines.					Speed in knots per hour.	Indicated horsepower.											
		To make one knot.		Per minute.				Starboard engine.					Port engine.					Total both main engines.	Auxiliaries in operation (estimated).	Total all machinery.
		Starboard.	Port.	Starboard.	Port.	Mean.		H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.	H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.			
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
	m. s.																			
1-S	9-57.8	610.9	588	61.3	59	60.65	6.022	34.84	32.41	10.94	10.94	89.13	32.42	29.16	8.78	9.66	80.02	169.15	25.48	194.63
2-N	8-15.91	625.6	611.6	75.6	74	74.8	7.26	63.31	56.41	16.87	12.37	148.96	54.68	43.88	14.31	17.07	129.94	274.9	23.46	298.36
3-S	6-33.1	591.6	586.2	90.1	89.3	89.7	9.142	96.1	96.17	28.82	19.44	240.53	91.36	75.02	25.91	29.89	222.18	462.71	25.58	488.29
4-N	6-04.74	638.1	635.5	104.9	104.5	104.7	9.871	148.06	145 67	49.16	35.9	378.79	138.59	129.1	48.2	42.76	358.65	737.44	29.31	766.75
5-S	4-59.44	613.5	608.1	122.9	121.9	122.4	12.024	224.69	242.93	90.05	74.97	632.64	207.85	210.84	76.17	75.27	570.13	1,202.77	34.11	1,236.88
6-N	4-41.48	649.4	653.5	138.4	139.3	138.8	12.789	320.94	324.18	129.21	111.19	885.52	300.85	309.77	124.35	129.53	864.5	1,750.02	36.8	1,786.82
7-S	4-21.12	626.5	629	143.9	144.5	144.2	13.788	365.31	366.92	156 29	156.29	1,044.81	355.88	357.03	150.49	155.86	1,019.26	2,064.07	47.43	2,111.5
8-N	4-20.39	666.1	667.1	153.5	153.7	153.6	13.825	503.11	474.09	183.84	189 55	1,350.59	419.3	425.33	177.22	182.94	1,204.79	2,555 38	48.65	2,604.03
9-S	3-54.38	639.8	639.2	163.8	163.6	163.7	15.356	550.82	586.84	231.51	220.55	1,589.72	548.6	557.83	237.31	237.31	1,581.05	3,170.77	71.93	3,242.7
10-N	3-55.16	677.4	691	172.8	176.3	174.6	15.306	641.66	713.02	300.79	278.94	1,934.41	647.96	688.25	305.05	308.2	1,949.46	3,883.87	70.79	3,954.66
11-S	3-32.95	665.1	666.6	187.4	187.8	187.6	16.901	809.46	926.06	405.66	414.04	2,555.22	800.53	909.5	408.77	412.13	2,530.93	5,086.15	65.25	5,151.4
12-N	3-44.81	694.8	717.9	185.4	191.6	188.5	16.014	772.74	911.6	402.72	426.17	2,513.23	675 17	946.83	560.14	534.49	2,716.63	5,229.86	115.54	5,345.4
13-S	3-35.63	660.5	663.1	183.8	184.5	184.1	16.698	612.86	876.5	512.73	408.82	2,410.91	594.23	927.39	528.41	511.25	2,363.28	4,774.19	97.96	4,872.15

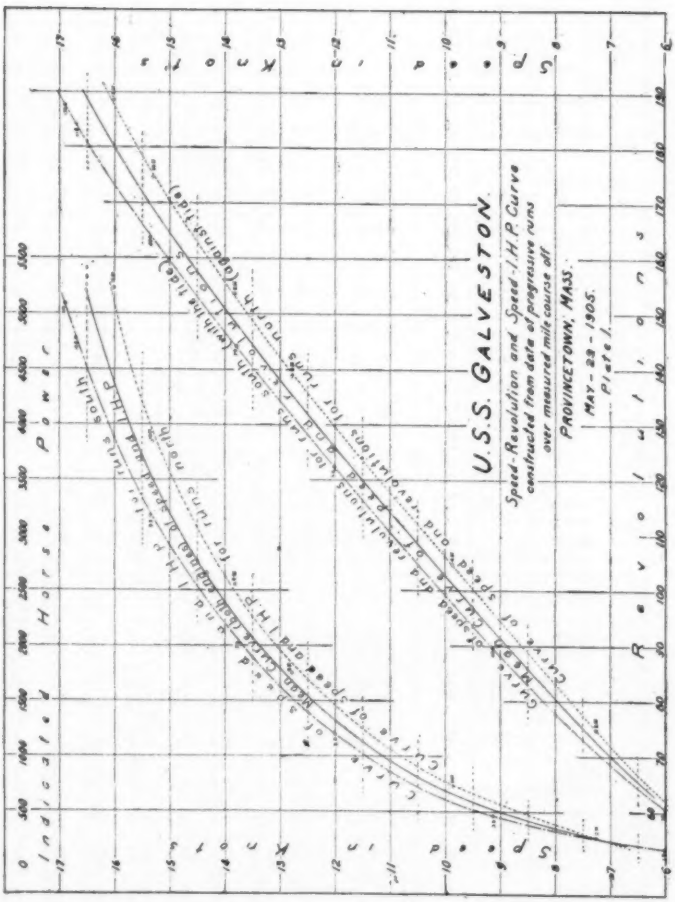
Table IV.—U. S. S. *GALVESTON.*

Data from Progressive Runs over Measured-Mile Course off

Table IV.—U. S. S. *GALVESTON.*

Data from Progressive Runs over Measured-Mile Course off Provincetown, Mass., May 29, 1905.

Number of run.	Time at beginning of run (approximate).		Steam pressures.				Mean effective pressures.					Cut off in cylinders from beginning.				Temperatures.					Revolutions or double strokes.							
			In boilers above atmosphere.	At engine above atmosphere.	H.P. steam chest, absolute.	In first receiver, absolute.	In second receiver, absolute.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to low pressure.	Vacuum, inches.	H.P.	I.P.	Forward L.P.	After L.P.	Outside air.	Engine-room working platform.	Fire-room working level.	Injection.	Discharge.	Hotwell.	Feed.	Circulating pump.	Main feed pumps.	Hotwell pump.	Water-service pump.	Bilge pump.	Aux. cond'r. pump.	Fire-room blowers.	Air pressure, inches water.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33
1–s	2:18	Sr'b'd	208	207	35	13	6	15	5:35	1.2	1.2	4.887	27	Cut off at .743.				75	89	107	50	69	100	82	110	14	12	57	50	34	...	...
		Port	208	207.5	33	13.5	4.5	14.5	5	1	1.1	4.557	27						93		50	61	82	200	160	9	11	38			...	...
2–n	2:45	Sr'b'd	208	207	47	17	6.5	22.1	7.55	1.5	1.1	6.621	27.2					75	89	110	50	70	112	185	110	12	9	58	30	35	...	...
		Port	208	207.5	42	15	5	19.5	6	1.3	1.55	5.901	27						93		50	62	80	204	152	9	12	48			...	...
3–s	3:12	Sr'b'd	210	207	60	21	7	28.15	10.8	2.15	1.45	8.971	27.5					74	90	108	50	70	112	200	120	12	11	56	32	32	...	...
		Port	210	208.5	58	20	6	27	8.5	1.95	2.25	8.361	27.5						93		50	64	74	194	156	12	24				...	...
4–n	3:33	Sr'b'd	215	207	75	27	9	37.25	14.05	3.15	2.3	12.135	28					74	90	107	50	80	112	205	120	16	12	57	30	34	...	...
		Port	214	209.5	74	26	7	35	12.5	3.1	2.75	11.534	27.5						90		50	67	72	186	156	15	26		48		...	...
5–s	3:55	Sr'b'd	245	249	92	37	11	48.25	20	4.925	4.1	17.302	28					73	90	110	50	80	98	215	120	20	12	63	42	36	...	...
		Port	246	247.5	96	33	9	45	17.5	4.2	4.15	15.718	27.5						90		50	70	67	180	168	16	24		56		170	...
6–n	4:13	Sr'b'd	255	228	135	40	13	61.2	23.7	6.875	5.4	21.502	28					73	90	103	50	87	92	215	130	20	14	64	38	36	170	...
		Port	255	232.5	122	42	12.5	57	22.5	6	6.25	20.856	27.5						90		50	78	72	180	168	19	25		56		160	...
7–s	4:31	Sr'b'd	270	249	157	55	15	67	25.8	7.3	7.3	24.4	28					72	87	99	50	85	90	210	130	38	18	64	36	40	170	...
		Port	270	261.5	137	49	14.5	65	25	7	7.25	23.708	27.5						90		50	78	71	194	159	18	28		60		205	...
8–n	4:49	Sr'b'd	260	249	167	57	22	86.5	31.25	8.05	8.3	29.57	28					72	90	92	50	85	92	190	135	38	34	64	40	40	155	...
		Port	260	241.5	151	53	15.5	72	28	7.75	8	26.344	27.5						90		50	78	71	194	152	30	23		52		170	...
9–s	5:04	Sr'b'd	265	249	190	65	23	88.75	36.25	9.5	9.05	28.615	28					72	90	96	50	85	90	180	145	50	40	64	50	40	215	...
		Port	265	257.5	185	64	19.5	88.5	34.5	9.75	9.75	28.48	27.5						90		50	84	69	170	152	34	48		54		180	...
10–n	5:26	Sr'b'd	275	259	220	75	25	98	41.75	11.7	10.85	37.62	27.5					72	90	92	50	95	95	185	145	44	40	64	50	36	220	...
		Port	275	250.5	215	78	25	97	39.5	11.63	11.75	37.160	27						90		50	85	74	169	162	40	32		60		340	...
11–s	5:40	Sr'b'd	280	256	262	100	30	114	50	14.55	14.85	45.82	27					72	90	87	50	110	108	180	160	40	50	64	50	36	370	...
		Port	280	251.5	237	97	30.5	122.5	49	14.03	14.75	45.89	26.5						90		50	100	81	184	172	33	46		56		395	...
12–n	5:52	Sr'b'd	280	254	260	98	30	110	49.75	14.60	15.45	45.555	27.5					71	90	93	50	95	95	180	180	64	50	61	50	36	489	...
		Port	280	240.5	243	105	37	93	39	19.65	18.75	47.65	26.5						90		49	91	86	192	196	60	26		54		471	2
13–s	6:06	Sr'b'd	260	259	265	102	33	88	48.25	18.25	14.95	44.08	27.5					71	86	90	50	95	95	180	180	60	50	60	50	40	433	...
		Port	260	234.5	235	93	36	85	40	19.25	18.63	43.05	26.5						90		48	94	84	170	180	56	32		52		507	2
																															421	


U.S.S. GALVESTON.
Speed-Revolution and Speed-I.H.P. Curve
constructed from data of progressive runs
over measured mile course off
PROVINCETOWN, MASS.
MAY - 23 - 1905.
Plate I.
Revolutions
Horse Power
Speed
Curve of Speed and Revolutions for runs North (Boston bound)
Curve of Speed and Revolutions for runs South (Gloucester bound)
Curve of Mean Speed and I.H.P. for runs North
Curve of Mean Speed and I.H.P. for runs South

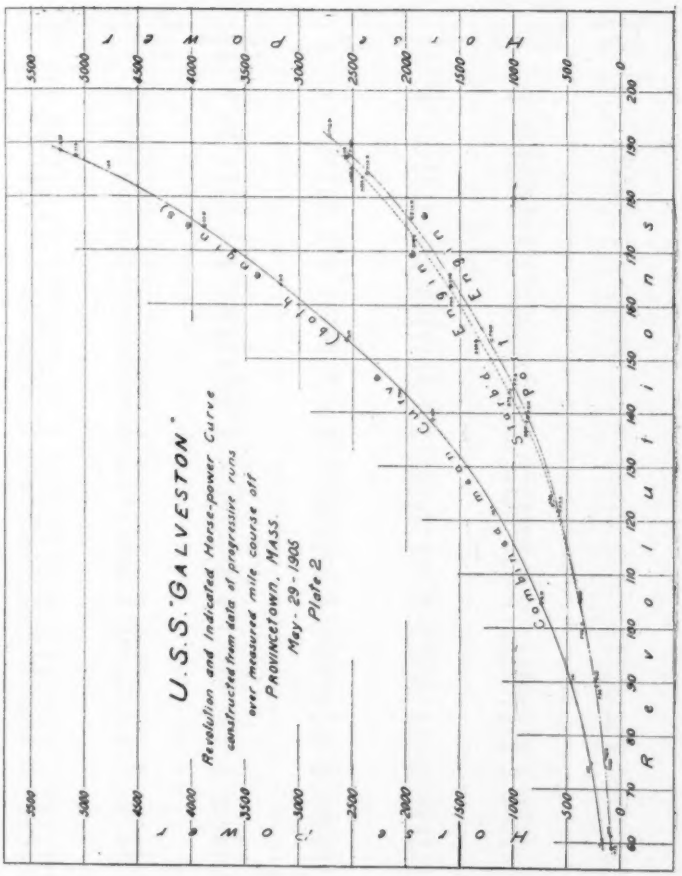

U.S.S. "GALVESTON"
Revolution and Indicated Horse-power Curve
constructed from data of progressive runs
over measured mile course off
PROVINCETOWN, MASS.
May - 29 - 1905
Plate 2
Revolutions
Indicated Horse-power
Speed

Table V.—U. S. S. *GALVESTON.*

Data deduced from Progressive Runs over Measured-Mile Course off Provincetown, Mass., May 29, 1905.

Speed in knots per hour.	Revolutions per minute.	Indicated horsepower.			Slip of propeller in per cent. of its own speed.	Constant, R ÷ S.
		Starboard engine.	Port engine.	Total both engines.		
6	**61.5**	**90**	**90**	**180**	**15.55**	**10.25**
6.1	62	91	91	182	14.83	10.162
6.2	63	94	94	188	14.80	10.162
6.3	63.8	96	96	192	14.51	10.127
6.4	64.7	100	100	200	14.36	10.110
6.5	65.6	102	102	204	14.22	10.093
6 6	66.5	105	105	210	14.08	10.075
6.7	67.4	108	108	216	13.94	10.060
6.8	68.3	111	111	222	13.80	10.045
6.9	69.2	115	115	230	13.68	10.030
7	**70.2**	**120**	**120**	**240**	**13.68**	**10.028**
7.1	71.1	123	123	246	13.55	10.015
7.2	72.1	126	126	252	13.55	10.015
7.3	73 1	130	130	260	13.55	10.013
7.4	74.1	135	135	270	13.54	10.013
7.5	75.2	140	140	280	13.66	10.026
7.6	76.2	145	145	290	13.66	10.027
7 7	77.3	150	150	300	13.76	10.038
7.8	78 3	158	158	316	13.76	10.038
7 9	79.4	163	163	326	13.87	10.050
8	**80.5**	**169**	**169**	**338**	**13.96**	**10.082**
8.1	81.5	175	175	350	13.95	10.060
8.2	82.6	180	180	360	14.06	10.072
8.3	83.7	185	185	370	14.15	10.083
8.4	84 8	195	195	390	14.24	10.095
8.5	85.8	202	202	404	14.22	10 095
8 6	87	210	210	420	14.43	10.117
8.7	88.1	217	217	434	14 51	10.127
8.8	89.2	226	226	452	14.59	10.135
8.9	90.2	234	234	468	14.58	10.135
9	**91.4**	**242**	**242**	**484**	**14.76**	**10.155**
9.1	92.5	252	252	504	14 84	10.165
9.2	93 6	260	260	520	14.91	10.174
9.3	94.8	270	270	540	15.07	10.193
9 4	95 8	280	280	560	15.07	10.190
9.5	97	290	290	580	15.22	10.210
9.6	98.1	302	302	604	15.28	10 220
9.7	99.3	315	315	630	15.43	10.236
9.8	100.4	330	330	660	15.50	10.245
9.9	101.6	337	337	674	15.65	10.260
10	**102.7**	**350**	**350**	**700**	**15.71**	**10.27**
10.1	103.8	363	360	725	15.77	10.280
10.2	105	380	375	755	15.90	10.295
10.3	106 2	390	385	775	16.12	10.310
10.4	107.2	405	395	800	16.01	10.310
10.5	108.5	420	410	830	16.22	10.335
10.6	109.5	430	420	850	16.20	10.330
10.7	110.6	445	435	880	16.25	10.337
10.8	111.9	460	450	910	16.46	10.362
10.9	113	475	460	935	16.50	10.367

Table V.—Continued.

Speed in knots per hour.	Revolutions per minute.	Indicated horsepower.			Slip of propeller in per cent. of its own speed.	Constant, R ÷ S.
		Starboard engine.	Port engine.	Total both engines.		
11	**114.3**	**495**	**475**	**970**	**16.69**	**10 390**
11.1	115.3	505	485	990	16.66	10.390
11.2	116 5	520	500	1,020	16.77	10.390
11.3	117.7	535	515	1,050	16.88	10.420
11.4	118.4	550	530	1,080	16 65	10.386
11.5	120	575	540	1,115	17 04	10.435
11 6	121.2	585	565	1,150	17.15	10.450
11.7	122 3	605	580	1,185	17.18	10.450
11.8	123.5	620	600	1,220	17 29	10.465
11.9	124.6	635	615	1,250	17.32	10 470
12	**125.9**	**655**	**630**	**1,285**	**17.48**	**10.490**
12.1	127.1	675	650	1,325	17.58	10.505
12.2	128.3	700	670	1,370	17.68	10.518
12.3	129.5	720	690	1,410	17 78	10.530
12.4	130 7	745	710	1,455	17.86	10.540
12.5	132	770	735	1,505	18.01	10 560
12.6	133.2	790	760	1,550	18.11	10.570
12 7	134.4	820	780	1,600	18.20	10.580
12 8	135 6	845	805	1,650	18 28	10.593
12.9	136 9	880	825	1,705	18.42	10.613
13	**138.2**	**905**	**850**	**1,755**	**18.58**	**10.630**
13.1	139.5	940	873	1,815	18.70	10.650
13.2	140.6	970	900	1,870	18.72	19 650
13 3	142	1,000	935	1 935	18.92	10.680
13.4	143 2	1,020	970	1,990	19.00	10.685
13.5	144 4	1,060	990	2,050	19 07	10.697
13.6	145.7	1,095	1,020	2,115	19.20	10.715
13.7	147	1,130	1,055	2,185	19 32	10 727
13 8	148.3	1,165	1,090	2,255	19.45	10.745
13 9	149.6	1,205	1,120	2,325	19 57	10.760
14	**150.8**	**1,240**	**1,150**	**2,390**	**19.64**	**10.770**
14 1	152.2	1,270	1,200	2,470	19.81	10.795
14.2	153 5	1,315	1,230	2,545	19 92	10.810
14.3	154.8	1,350	1,270	2,620	20 03	10.825
14.4	156.2	1,390	1,310	2,700	20.19	10.850
14.5	157.5	1,430	1,355	2,785	20.31	10.860
14 6	159	1,470	1,400	2,870	20.51	10.890
14.7	160.3	1,515	1,435	2,950	20.61	10.905
14.8	161.7	1,555	1,485	3,040	20.76	10.925
14.9	163.1	1,600	1,525	3,125	20 91	10.945
15	**164.5**	**1,645**	**1,570**	**3,215**	**21.06**	**10.967**
15.1	166	1,700	1,615	3,315	21.25	19.993
15.2	167.4	1,745	1,665	3,410	11.38	11.013
15.3	168.8	1,790	1,710	3,500	21.53	11.033
15.4	170.3	1,840	1,760	3,600	21.71	11.060
15.5	171.9	1,900	1,815	3.715	21 96	11.090
15.6	173.5	1,960	1,875	3.835	22.17	11.120
15.7	175	2,020	1,930	3,950	22.33	11 150
15.8	176 5	2,080	1,990	4,070	22.51	11.170
15.9	178.2	2,150	2,060	4,210	22.76	11.210
16	**180**	**2,225**	**2,130**	**4,355**	**23.06**	**11.250**
16.1	181.5	2,295	2,200	4.495	23.22	11.270
16.2	183.3	2,380	2,285	4,665	23.50	11.315
16.3	184	2,470	2,370	4,840	23.73	11.350
16.4	186.9	2,570	2,470	5,040	24 04	11.397
16.5	188.4	2,645	2,555	5,200	24.19	11 420
16.6	190.6	2,770	2,680	5.450	24 59	11.482

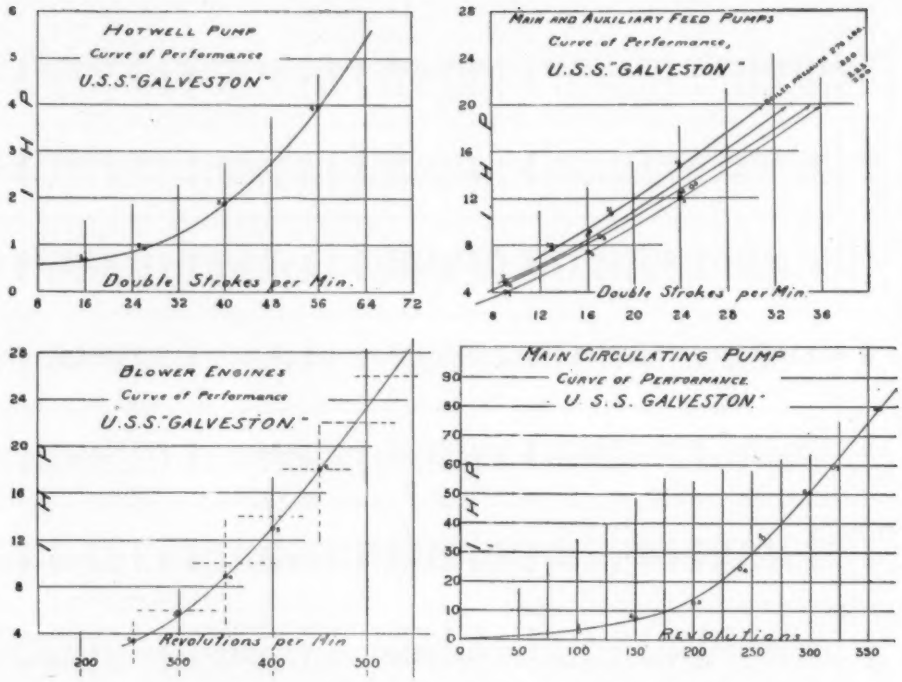


Plate III.—PERFORMANCE CURVES AUXILIARY MACHINERY.

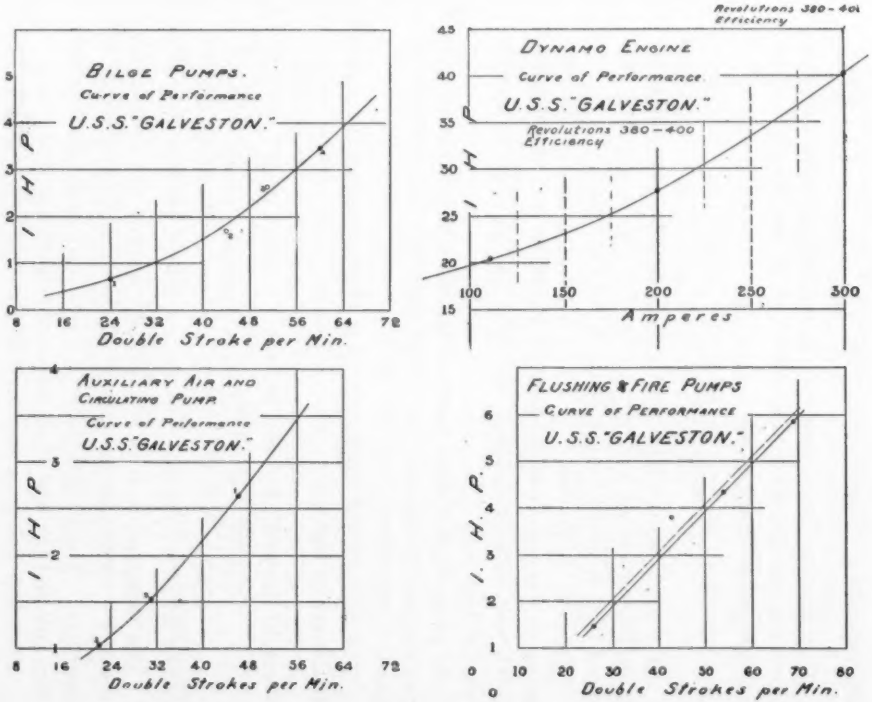


Plate IV.—PERFORMANCE CURVES AUXILIARY MACHINERY.

The draught at beginning was as follows:

Forward, feet and inches,	15–10½
Aft, feet and inches,	16– 4
Mean, feet and inches,	16– 1¼
Corresponding displacement, tons,	3,295

The time on the course was taken independently by four observers, and the revolutions checked by not less than three counters, of different types.

FOUR-HOURS' OFFICIAL TRIAL.

At 7·30 A. M. on May 30th the *Galveston* got under way from Provincetown and put to sea for the four-hours' full-speed trial required by the contract for the vessel.

The trial commenced at 10·30 A. M., the ship at the time being on a course N.N.W. ½ W. in a smooth sea; the weather conditions were good, calms and light airs prevailing throughout the trial.

At 12·30 P. M. a wide turn was made to the northward with two degrees of helm, and the vessel headed in toward Provincetown.

The draught and displacement at the beginning and end of trial were as follows:

	Beginning.	*End.*
Draught, forward, feet and inches,	15– 9	15– 6
aft, feet and inches,	16– 3½	16– 3
mean, feet and inches,	16– 0¼	15–10½
Displacement, tons,	3,280	3,230

The main and auxiliary machinery worked in a satisfactory manner; there was no undue vibration and no heating of bearings. The boilers steamed freely during the standardization run and the first three hours of the official trial.

The ship being in commission, the trial was run with the regular crew, a number of whom had only recently joined. The official trial was the first full-power run made.

The firing of the furnaces was controlled by an electric indicator; as the furnace numbers were shown, coal was thrown

on the front of the grate and immediately afterwards the coals in an equal number of furnaces were leveled with a devil's claw or hoe. Slice bars were used occasionally through the slicing doors.

Steam pressure at the engine was limited to 250 pounds, but the safety valves on boilers were permitted to be set at 289 pounds. The air pressure in fire rooms was limited to two inches of water.

The mean average revolutions for the four hours corresponded to a speed of 16.406 knots per hour, while for the first three hours this amounted to 16.54 knots.

The following is a summary of the data recorded.

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Steam pressure at boilers (per gauge), pounds	263	
engines	247	248
H.P. steam chest (gauge), pounds	240	238
1st receiver (absolute), pounds	96.6	102
2d receiver (absolute), pounds	31.7	31.5
Vacuum in condensers, inches of mercury	27.5	27

Temperatures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Injection, degrees	45.4	45.4
Discharge, degrees	85.7	83
Hotwell, degrees	89.6	72
Feed-water, degrees	186	183
Engine room, upper platform, degrees	85	91
working platform, degrees	85	80
Firerooms, working level, degrees	83.1	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Main engines (maximum for 15 minutes)	191.46	192.9
(minimum for 15 minutes)	174	176.1
(mean for trial)	185.89	187.96
average for both	186.925	
Pumps, main, air	Attached.	
circulating	183	184
feed (forward and after)	34.4	31
hotwell	49.4	26
bilge	66	57
water service	...	59
auxiliary, feed	25.7	25
Blower engines, starboard, forward	491	

	Starboard.	*Port.*
Blower engines, port, forward	493	
after	440	
Speed of ship, in knots per hour	16.406	
Slip of propeller, in per cent. of its own speed, based on mean pitch and average speed	23.53	24.45
Air pressure in firerooms, inches of water	2	

Mean effective pressures in cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods*).

	Starboard.	*Port.*
Main engines, H.P. cylinder	110.13	105.69
I.P. cylinder	48.09	49.875
F.L.P. cylinder	14.73	14.825
A.L.P. cylinder	14.78	14.824
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	44.755	44.853

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	775.71	752.7
I.P. cylinder	883.50	926.53
F.L.P. cylinder	407.38	414.58
A.L.P. cylinder	408.76	414.55
Total	2,475.35	2,508.36
both engines	**4983.71**	

AUXILIARIES.

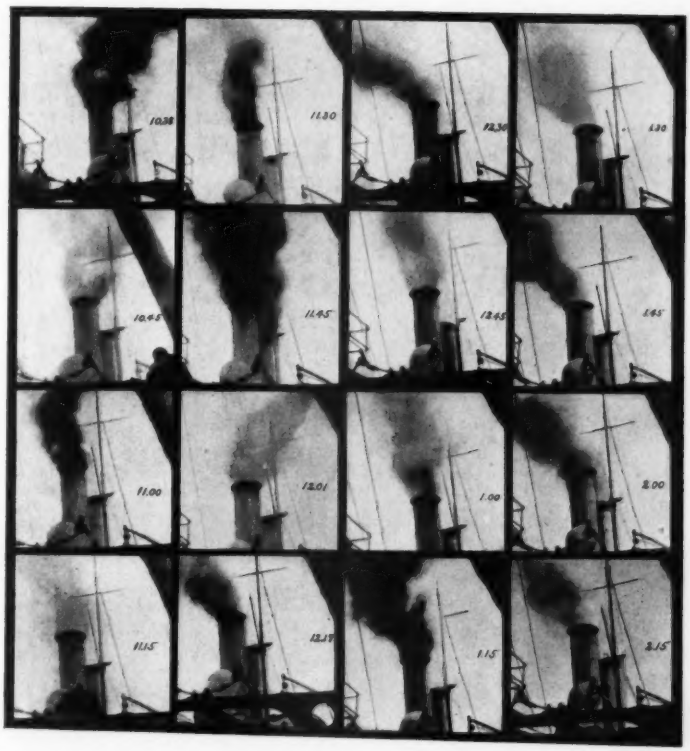
Circulating pumps, main	11.1	11.2
Feed pumps (main and auxiliary)	67.42	
Hotwell pumps	2.85	1
Bilge pumps	4.13	3.13
Water-service pump	4.84	
Auxiliary condenser pumps	1.17	
Blower engines (forced draft)	62.25	
Dynamo engine (one in operation)	25.67	
Total	**194.76**	
all machinery in operation	**5,178.47**	

COAL.

Kind and quality	Pocahontas, run of mine.
Pounds per hour	10,022

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	17.26
heating surface	.3923
Pounds of coal per I.H.P. per hour (main engines, circulating, feed and hotwell pumps)	1.974
all machinery in operation,	1.935
Pounds of coal per square foot of grate surface	33.41
heating surface	.758
Cooling surface (main condenser), square feet per (total) I.H.P.	1.238
Heating surface (boilers), square feet per (total) I.H.P.	2.549



PHOTOGRAPHS OF SMOKE TAKEN DURING TRIAL, U. S. S. "GALVESTON."

Table VI.—U. S. S. *GALVESTON.*

Performance on 24-hour Endurance Trial, May 31 to June 1, 1905.

Table VI.—U. S. S. *GALVESTON.*

Performance on 24-hour Endurance Trial, May 31 to June 1, 1905.

.....	1st period, 11 A.M. to 11 P.M. May 31, 1905.	2d period, 11 P.M. May 31, to 11 A.M. June 1, 1905.	1st and 2d periods combined, 11 A.M. May 31 to 11 A.M. June 1, 1905.
	Starboard. *Port.*	*Starboard.* *Port.*	*Starboard.* *Port.*
Number of hours	12	12	24
Draft	Natural.	Forced.	Natural and forced.
Mean revolutions of engines	152.97 153.1	160.06 160	156.5 156.5
both engines	153.03	160.03	156.5
Steam pressure in boilers above atmosphere, pounds	233.58	226.25	229.4
H.P. cylinder above atmosphere, pounds	228.58 226.5	224.17 221.66	226.4 224.1
first receiver, absolute, pounds	61.78 59.6	69.87 65.95	65.8 62.8
second receiver, absolute, pounds	23.95 22.1	28.7 24.8	26.3 23.45
Steam cut-off in H.P. cylinder, inch	0.52	0.52	0.52
I.P. cylinder, inch	0.56	0.56	0.56
L.P. cylinders, inch	0.5	0.5	0.5
Vacuum, inches	27.58 27.66	27.54 26.5	27.6 27.64
Temperature, fireroom, degrees Fahrenheit	125.25	97	108.3
injection, degrees Fahrenheit	49.92 49.66	53.83 53.33	51.9 51.5
discharge, degrees Fahrenheit	89.66 80	91.66 83.66	90.7 81.8
feed water, degrees Fahrenheit	163.5 156	156.9 159.33	160.2 157.7
Revolutions of air pumps per minute	Attached.	Attached.	Attached.
circulating pumps per minute	75 115	116.1 119.6	95.5 117.4
Grate surface in use, square feet	300	300	300
I.H.P. main engines	1,234.9 1,280.29	1,385.4 1,414.9	1,347.59 1,310.15
both engines	2,515.19	2,800.3	2,657.74
auxiliaries, estimated	160	160	160
Aggregate I.H.P.	2,675.19	2,960.3	2,817.74
Pounds of coal per hour	5,167.75	5,692.5	5,430.12
per square foot of grate in use	17.226	18.975	18.1
I.H.P., main engines	2.054	2.032	2.043
and auxiliaries	1.931	1.923	1.927
Mean draught at beginning of trial, feet and inches	15–10½	15–10½	15–10½
Displacement, tons	3,230	3,230	3,230
Mean speed by patent log in knots per hour	14.49	15.13	14.81
speed curve in knots per hour	14.14	14.7	14.44
Slip of screw, per cent	16.77	17.26	16.89
Average air pressure, inches of water	None.	0.375	0.1875

A METHOD OF CALCULATING THE COAL ENDURANCE AND STEAMING RADIUS OF WAR VESSELS UNDER VARIOUS CONDITIONS.

By Lieut. Commander B. C. Bryan, U. S. Navy, Member.

The difficulty of procuring a supply of coal in time of war, and the necessity of steaming long distances without coaling, makes it important that a commanding officer should know the probable coal consumption and steaming radius of the vessel under his command.

The writer, a few years ago, spent considerable time collecting and tabulating the data at hand concerning U. S. naval vessels, with the object of forming a guide to enable the senior engineers and commanding officers of such vessels to tell what results might be expected under various conditions.—(See Journal of American Society of Naval Engineers, Vol. XIII, p. 50.)

Other articles upon this subject have appeared in this journal, both before and after the above compilation was published.—(See "Economic Marine Propulsion," John Lowe, Vol. III, p. 218; "Coal Endurance and Machinery of New Cruisers," I. N. Hollis, Vol. IV, p. 637; "Economic Speed and Power of War Vessels," &c., Vol. V., p. 66; "Speed and Coal-Consumption Curves," D. S. Mahony, Vol. XV, p. 366.)

The subject has been very ably treated in the above articles, and it is only with the idea of presenting in a practical way a method of tabulating and using the data collected that this article is written.

The Bureau of Steam Engineering has recognized the importance of collecting the necessary data from long runs of all vessels, and to this end has added a "Coal Endurance Form" in the back of the steam log, to be filled in for all continuous runs of seventy-two hours' duration or more. Although the results of a number of such runs are now on file, many of them are so incomplete or contain such glaring inaccuracies that it

is often impossible to account for the erratic performances tabulated. In order to collect such data so that it may be of value, care and co-operation from those concerned is required; the coal and revolutions must be accurately recorded in the steaming log, together with all data concerning steam-using appliances, the speed through the water, weather conditions, trim and all conditions influencing the results should be accurately recorded in the deck log; thus co-operation between the navigating, engineer and deck officers is essentially necessary.

The problem presented to a commanding officer or senior engineer of a vessel is as follows: "The ship having come into port with the bottom more or less fouled, requiring a certain number of revolutions, exceeding the number required with a clean bottom, to make a certain speed, what would be the probable coal consumption at any other speed under the same conditions and what the most economical speed?" To answer this, in a practical way, is, as above stated, the object of this article.

From extensive investigation of the data on file for U. S. naval war vessels, it is believed that the coal consumption follows very closely the formula deduced by Chief Engineer John Lowe, U. S. Navy (now Rear Admiral, Retired), and given by him in his article in the JOURNAL of the Society, Vol. III, pp. 218–221, as follows: $Y=K+CR^3$, when

Y = total coal used for all purposes in tons per day;

R = average revolutions per minute of the main engines;

K = a constant, representing the coal in tons per day necessary for auxiliary purposes plus that necessary for keeping the main engine warmed when making zero revolutions;

C = a quantity, constant for given conditions.

Determination of the constant K should be made by a long trial or trials at the dock, the longer the better, with the auxiliaries in use as near as possible the average under usual conditions and the engines running as slowly as it is possible to run them. This will give the value of K under the usual conditions, and it may be corrected for other conditions by observations of the additional amount of coal necessary to produce those conditions by trials made while lying in port.

The valve of C should be determined by the results of one or more long runs, under average conditions if possible. The results of all long runs of the vessel while in commission should be carefully tabulated on forms similar to Table I (or the coal consumption form to be found in back of steam log), and from these results C may be corrected from time to time, and other curves drawn to suit particular or peculiar conditions.

If no trial has been made to determine the value of K, an approximation may be obtained by selecting a number of actual runs and taking the value of R at several different points and the corresponding value of Y (the total coal per day), then assuming different values of C and subtracting the values of CR^3 from the corresponding values of Y until all values of the differences become the same or vary the least, the average difference then is an approximation for K. In doing this care should be taken that the data has been under normal conditions and accurately taken, and it is best to take an average of several runs for each value of Y and R. In case the value of K is found to be less than the average amount of coal used in port or less than the value of N_2 as found by the method given by Prof. I. N. Hollis in his article on "Economical Speed and Coal Endurance of War Vessels," &c., published in the JOURNAL, Vol. V, pp. 79 and 80, it may be taken for granted that the data is inaccurate.

For finding the relation of speed to revolutions a progressive trial should be made over a measured course and the usual data accurately recorded. The vessel should be clean when this trial is made, and the results will be the best that may be expected.

The value of the slip for each speed is now determined from these results.

To obtain the corresponding revolutions for any speed under different conditions of foulness, it is assumed that the slip will vary in the same proportion for each speed, that is, that if it is found that with a certain speed S when entering a port a certain number of revolutions R^1 is required giving a slip of Φ^1, the number of revolutions for the same speed with a

clean bottom being R and the slip being Φ, then $\frac{\Phi^1}{\Phi}$ is a constant for all speeds, under the same conditions.

We then have $\dfrac{P \times R - S \times \frac{6{,}080}{60}}{P \times R} \times 100 = \Phi$, or

$$P \times R\,(100 - \Phi) = S \times 10{,}133.3,$$

Similarly, $P \times R^1\,(100 - \Phi^1) = S \times 10{,}133.3$, or

$$P \times R\,(100 - \Phi) = P \times R^1\,(100 - \Phi^1)$$

$$R^1 = R\left\{\frac{100 - \Phi}{100 - \Phi^1}\right\}.$$

From this the number of revolutions for any speed may be calculated for the existing conditions, the number of revolutions for one speed being known for the same conditions, and the equation of coal consumption $Y = K + CR^3$ will immediately allow the calculation of the probable coal consumption at the new speed.

The method recommended to be pursued is as follows:

Records of all long trips should be kept on blanks similar to Table I. The coal should be accurately recorded, and speed should be that which is made by the ship through the water; any error in the log should be allowed for and current should be eliminated. Notes should be made on each run of any occurrence or the use of any extraordinary amount of machinery, extra water distilled, or any use of steam that would affect the result. After a number of runs have been recorded, if in the meantime the trials for finding K have not been made, the curve is to be fitted to the average of the results as described. In doing this it is well to take the average of several runs at about the same speed for each point, and to take at least four points, one being as low as possible and another as high as it is likely long runs will be made.

Judgment must be used in selecting the runs. Short runs, and particularly those made for obtaining speed records for a few hours with limited auxiliaries in use, should be avoided, also runs in squadron, when reserve power is necessarily re-

quired with corresponding extra coal consumption. From these results the curve of coal consumption may be laid down on revolutions, the easiest way to do this being probably to assume revolutions varying by 10 and find the value of Y from the equation $Y=K+CR^3$.

If a progressive trial has not been made, values of the speeds and revolutions for runs with a clean bottom should be selected; the length of the run is not of first importance if the speed and revolutions have been accurately recorded under normal conditions; the contractor's trial will usually give one point, and several others should be taken from which a curve of slip with a clean bottom may be laid down and the slip determined under these conditions for every speed, from which the corresponding revolutions may be calculated by the usual formula.

Curves of speed on revolutions may now be laid down for any increased value of the slip, assuming that the slip is increased in the same ratio for all speeds. Knowing the bunker capacity, curves of steaming radii for each value of the slip ratio may be added. As tables are considered more desirable by some people, such tables may be made out for each assumed value of the slip ratio.

To illustrate the method to be employed, the results as tabulated for the battleship *Kearsarge* and cruiser *Cleveland* have been laid down and tables made out for several values of the slip ratio, and are appended. To illustrate the use of the same, it is supposed that the *Kearsarge* came into port making 10 knots with 71 revolutions, and it is desired to know what the probable coal consumption would be under the same conditions going out at 12 knots. Following the 10-knot horizontal line on curve diagram we find it strikes 71 revolutions on the curve of 100 per cent. increased slip ratio; following this curve to 12 knots we find it will take 87.5 revolutions. Following down the ordinate to the coal consumption, we find it takes 125 tons per day, giving a steaming radius of 3,623 knots, or the same results may be found by consulting Table III, C.

Table I.—U. S. S. *KEARSARGE*.

Kind of Coal

Table I.—U. S. S. *KEARSARGE.*

Kind of Coal, Various. Bunker Capacity, 1,573 Tons.

Number of run.	Number of hours' duration.	Speed in knots per hour.	Average revolutions per minute.	Total coal per day.	Steaming radius.	Date.	Days out of dock.	Number boilers in use.	Average steam pressure.	I.H.P.	Draught of vessel. F.	Draught of vessel. A.	REMARKS.—Cruising ground. Force and direction of wind and sea. Quality of coal. Number of fires banked. And any unusual amount of auxiliary machinery in use or any occurrence that will affect result.
				Tons.	*Knots.*	*1901.*			*Pounds.*				
5	95	9.734	63.8	67.3	5,450	Dec. 28	24	3	...	1,927	...	...	Havana to Culebra.—Light breeze on bow. Sea smooth. New River coal.
						1902.							
6	96	7.842	51.9	53.8	5,509	March 5	90	4	...	1,194	...	...	Cienfuegos to Colon.—Light wind forward beam. Sea smooth. Pocahontas coal.
7	119	7.75	62.8	65.92	5,078	March 15	100	4	...	2,124	...	...	Colon to Trinidad.—Fresh breeze on bow. Moderate sea on bow. Pocahontas coal.
8	96	9.9	64.6	67.4	5,536	Nov. 17	19	4	...	2,245	...	...	Hampton Roads to Culebra.—Light breeze on bow. Medium sea abeam. Eureka coal; fair quality.
						1903.							
9	72	9.9	64.7	69.3	5,378	Feb. 12	105	5	...	2,210	...	...	Ponce to Galveston.—Light breeze on quarter. Moderate sea on quarter. George's Creek coal; fair.
10	72	10.9	74.2	87	4,719	Feb. 15	108	5	...	3,348	...	...	Ponce to Galveston.—Light breeze on quarter. Moderate sea on quarter. George's Creek coal; fair.
11	95	10.87	75.9	96.5	4,247	April 25	150	5	147	3,465	...	...	Pensacola to Cape Charles.—Wind variable. Smooth sea, to rough and variable. Poor Pocahontas coal.
12	165	12.1	78.5	98	4,656	June 6	16	5	145	3,555	...	...	Pensacola to Southampton.—Wind light on bow. Sea ahead moderate. Eureka coal; fair quality.
14	220	13.1	87.5	124.9	3,838	July 20	60	5	124	5,191	...	...	Southampton to Bar Harbor.—Wind and sea variable. Best English Admiralty Cardiff coal.
15	111	10.67	69	76.9	5,146	Dec. 4	15	4	156	2,689	...	...	Staten Island to Guantanamo, Cuba.—Eureka coal of good quality.
						1904.							
17	71	12	81	108.9	4,089	May 7	180	4–5	165	4,293	...	...	Pensacola to Guantanamo.—Good Pocahontas coal.
18	95	11	72	86.2	4,735	May 22	195	3	169	2,937	...	...	Guantanamo to Horta Fayal.—Good Pocahontas coal.
20	94	7.13	47.8	57	4,620	Aug. 6	270	3	160	1,188	...	...	Fiume, Austria, to Gibraltar.—Fair Imperial Navigation coal.
21	118	9.67	64.8	74.5	4,777	Aug. 15	279	4	160	2,550	...	...	Gibraltar to Horta Fayal.—Imperial Navigation and Ferndale coal.
22	194	10.28	68.9	73.8	5,168	Aug. 24	290	4	160	2,881	...	...	Horta Fayal to Newport, R. I.—Ferndale and Pocahontas coal.

Table II.—U. S. S. *KEARSARGE.*

Data record of actual runs.						Comparison of actual and calculated data.	
No. of run.	Number of hours.	Revolutions.		Coal per day.		Coal per day from equation.	Difference from actual results (column 3).
		For run.	Average.	For run.	Average.		
14	220	87.5	84.3	124.9	116.9	125.1	+0.2
17	71	81		108.9		107.1	−1.8
12	165	78.5		98		100.9	+2.9
11	95	75.9	75.1	96.5	91.9	94.8	−1.7
10	72	74.2		87		91.1	+4.1
18	95	72		86.2		86.5	+0.3
15	111	69		76.9		80.7	+3.8
22	194	68.9		73.8		80.5	+6.8
21	118	64.8		74.5		73.4	−1.1
9	72	64.7	65.5	69 3	70.7	73.2	+3.9
8	96	64.6		67.4		73	+5.6
5	95	63.8		67.3		71.7	+4.7
7	119	62.8		65.9		70.2	+4.3
6	96	51.9	50	53.8	55.4	56.2	+2.4
20	94	47.8		57		52.2	−4.8

R =	50	65.5	75.1	84.3	
R^3 =	125,000	263,372	423,565	599,078	
Y =	55.4	70.7	91.9	116.9	
c = .00013	16.2	34.2	55.1	77.9	
	39.2	36.5	36.8	39	
				36.8	
				36.5	
				39.2	
				4/151.5	
				37.875	

$Y=38+.00013\ R^3$.

Table III.—U. S. S. *KEARSARGE.*

Bunker Capacity, = 1,573 tons. Coal-consumption curve for natural draft, $I' = 38 + .00013\ K^3$.

A.—PROGRESSIVE TRIAL RESULTS.

Speed in knots	4	5	6	7	8	9	10	11	12	13	14	15	16
Slip, per cent	...	...	...	7.64	7.85	8.22	8.94	9.25	9.8	10.5	11.1	11.88	...
Revolutions per minute	...	...	...	44.2	51	57.5	64	71.2	78	85.3	92.5	100	...
Coal per day, tons	...	...	...	49.2	55.2	62.7	72.1	84.9	99.7	118.7	140.9	168	...
Steaming radius, knots	...	...	...	5,371	5,471	5,418	5,235	4,892	4,543	4,135	3,751	3,370	...

Most economical speed = 52.7 revolutions = 8.27 knots. Steaming radius = 5,477 knots.

B.—SLIP 80 PER CENT. GREATER THAN PROGRESSIVE TRIAL.

Speed in knots	4	5	6	7	8	9	10	11	12	13	14	15	16
Slip, per cent	...	...	...	11.46	11.74	12.33	13.2	13.87	14.7	15.75	16.65	17.82	...
Revolutions per minute	...	...	...	46.1	53.3	60.2	67.3	75	82.5	90.6	98.7	107.2	...
Coal per day, tons	...	...	...	50.7	57.7	66.4	77.6	92.8	111	134.7	163	...	...
Steaming radius, knots	...	...	...	5,213	5,234	5,116	4,865	4,475	4,081	3,643	3,243	...	...

Most economical speed = 7.91 knots. Steaming radius = 5,238 knots.

C.—SLIP 100 PER CENT. GREATER THAN PROGRESSIVE TRIAL.

Speed in knots	4	5	6	7	8	9	10	11	12	13	14	15	16
Slip, per cent	...	...	...	15.28	15.7	16.44	17.88	18.5	19.6	21	22.2	23.8	...
Revolutions per minute	...	...	...	48.2	55.8	63.2	71	79.3	87.5	96.6	105.8	115.6	...
Coal per day, tons	...	...	...	52.6	60.6	70.8	84.5	102.8	125.1	155.2	...	...	...
Steaming radius, knots	...	...	...	5,023	4,984	4,800	4,468	4,040	3,623	3,162	...	...	...

Most economical speed = 7.6 knots. Steaming radius = 5,033 knots.

Table IV.—U. S. S. *CLEVELAND.*

Kind of Coal, Various. Bunker Capacity, 668 Tons.

Number of run.	Number of hours duration.	Speed in knots per hour.	Average revolutions per minute	Total coal per day.	Steaming radius.	Date.	Days out of dock.	Number boilers in use.	Average steam pressure.	I.H.P.	Draught of vessel. F.	Draught of vessel. A.	REMARKS.—Cruising ground. Force and direction of wind and sea. Quality of coal. Number of fires banked. And any unusual amount of auxiliary machinery in use or any occurrence that will affect result.
				Tons.	*Knots.*	*1904.*			*Pounds.*				
1	95	11.7	121.9	32.32	5,825	May 7	180	4	160	1,200	...	...	Pensacola to Guantanamo.—Light breeze on bow. Sea smooth. Pocahontas coal.
2	59	9.93	103.75	25.82	6,165	May 15	188	3	190	650	...	...	Guantanamo to St. Thomas.—Light air on bow. Sea smooth. Pocahontas coal.
3	117	10.32	107.08	28.27	5,861	May 20	193	4	205	...	...	...	St. Thomas to Horta Fayal.—Light breeze on bow. Sea smooth to medium. Cory's Merthyr coal.
4	107	10.15	102.7	24.14	6,740	May 26	199	4	...	...	...	...	St. Thomas to Horta Fayal.—Light breeze on bow. Medium sea. Cory's Merthyr coal.
5	71	13.49	134.8	38.32	5,644	June 1	204	4	200	...	...	...	Horta Fayal to Tangiers.—Light breeze aft. Sea smooth to medium. Cory's Merthyr coal.
7	73	10.1	105	25.2	6,426	Aug. 2	270	4	200	820	...	...	Corfu, Greece, to Ville France.—Light breeze on bow. Sea smooth. Excellent Cardiff coal.
8	111	10.6	105	25.2	6,680	Aug. 10	278	4	200	820	...	...	Ville France to Smyrna.—Light breeze on quarter. Sea Smooth. Excellent Cardiff coal.
9	149	10.56	106	26.4	6,400	Aug. 18	286	3–2	200	840	...	...	Smyrna to Gibraltar.—Light breeze on quarter. Sea smooth. Excellent Cardiff, mixed with poor Ferndale coal.
10	72	11.9	126.3	33.09	5,878	Sept. 5	304	4	200	1,250	...	...	Gibraltar to Cherbourg, France.—Light, variable breeze. Moderate sea. Excellent Pocahontas coal.
11	73	11.58	131.1	36.63	5,076	Nov. 3	363	4	200	2,000	...	...	Southampton to Gibraltar.—Light wind abeam. Sea smooth. Excellent Pocahontas coal.
12	195	12.76	131	39.2	5,210	Dec. 5	395	4	200	2,000	...	...	Gibraltar to Bridgetown, Barbadoes.—Light breeze on quarter. Sea smooth. Pocahontas coal.

Table V.—U. S. S. *CLEVELAND.*

Data recorded on actual runs.						Comparison of actual and calculated data.	
No. of run.	No. of hours.	Revolutions. Average for run.	Revolutions. Average.	Coal per day. For run.	Coal per day. Average.	Coal per day from equation.	Difference from actual result.
5	71	134.8	132.3	38.32	38.05	39	+ 0.68
11	73	131.1		36.63		37	+ 0.36
12	195	131.1		39.2		37	− 2.2
10	72	126.3	124.1	33.09	32.7	34.6	+ 1.5
1	95	121.9		32.32		32.6	+ 0.28
3	117	107.08	104.9	28.27	25.84	27.7	− 0.43
9	149	106		26.4		26.4	± 0.
8	111	105		25.2		26.1	+ 0.8
7	73	105		25.2		26.1	+ 0.8
2	59	103.75		25.82		24.7	− 0.12
4	107	102.7		24.14		25.3	+ 1.16

R =	104.9	124.1	132.3
R^3 =	1,154,320	1,911,240	2,315,685
Y =	25.84	32.70	38.05
c = .00001			
cR^3 =	11.54	19.11	23.16
	14.30	13.59	14.89

Average for $K = 14.26$ Call $K = 14.5$

Coal consumption $Y = 14.5 + .00001R^3$

Table VI.—STEAMING RADIUS U. S. S. *CLEVELAND.*

Bunker Capacity, 668 tons. $V=14.5+.00001\ R^3$.

A.—PROGRESSIVE TRIAL, CLEAN BOTTOM.

Speed, knots	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Slip, per cent	...	...	15.35	14.85	14.6	14.55	14.8	15.4	16.2	17.2	18.4	19.9	22	...
Revolutions per minute	...	...	61.28	71.08	81	91.05	101.5	112.4	123.8	135.75	148.3	161.9	177.4	...
Total coal per day, tons	...	...	16.8	18.1	19.8	22	25	28.7	33.4	39.3	48.1	59.9	...	...
Steaming radius, knots	...	...	5,725	6,200	6,478	6,558	6,413	6,145	5,760	5,302	4,666	4,015	...	...

Most economical speed = 89.8 revolutions = 8.88 knots. Steaming radius = 6,560.

B.—SLIP 50 PER CENT. MORE THAN CLEAN BOTTOM.

Speed, knots	...	...	6	7	8	9	10	11	12	13	14	15	...	...
Slip, per cent	...	...	23.02	22.31	21.9	21.83	22.2	23.1	24.3	25.8	27.6	...	...	...
Revolutions per minute	...	...	67.4	77.9	88.6	99.5	111.1	123.7	137	151.5	167.2	...	...	...
Total coal per day, tons	...	...	17.6	19.2	21.5	24.4	28.2	33.4	40.2	49.2	...	...	...	...
Steaming radius, knots	...	...	5,466	5,845	5,966	5,914	5,686	5,280	4,785	4,236	...	...	...	...

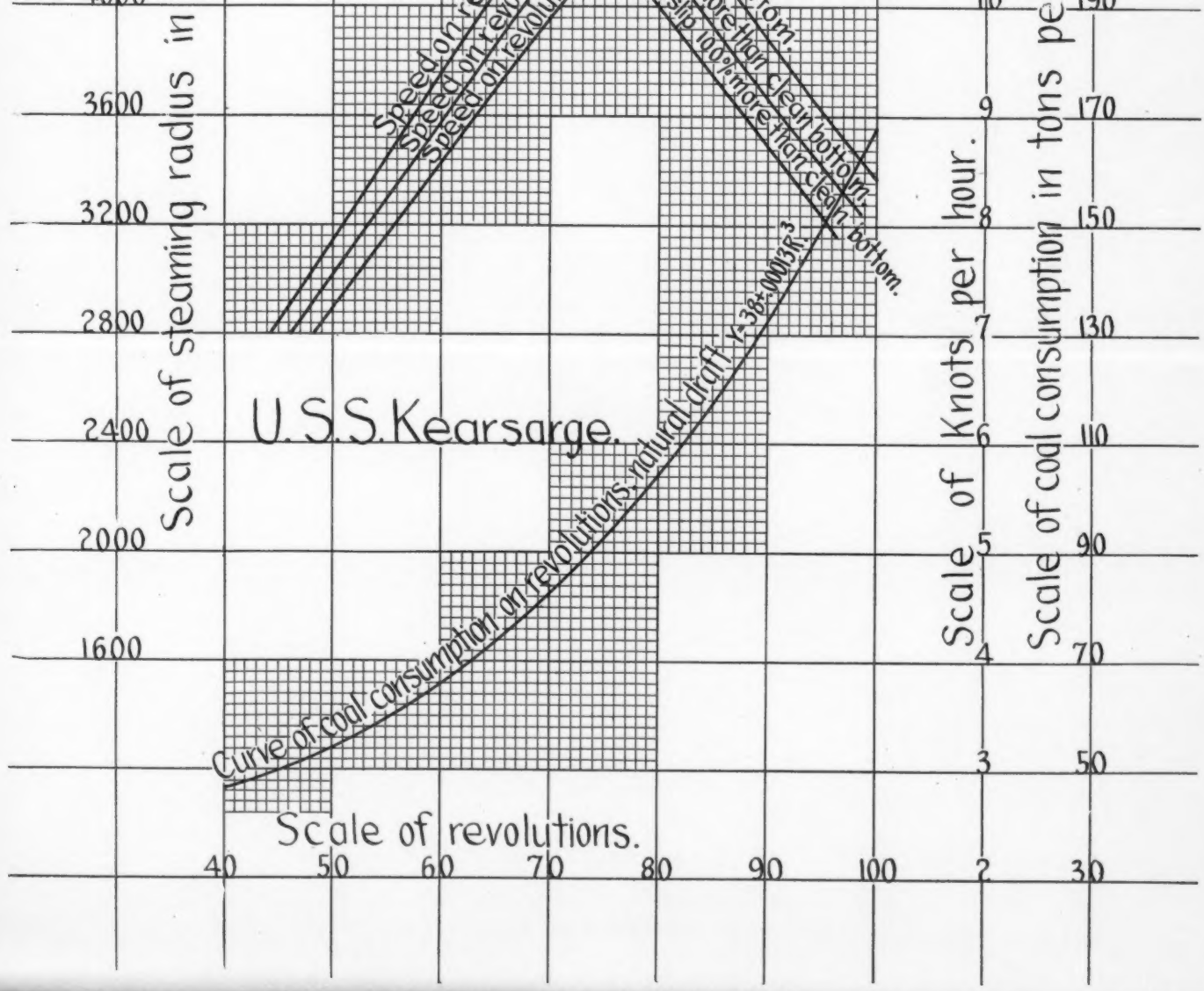
Most economical speed = 8.1 knots. Steaming radius = 5,984.

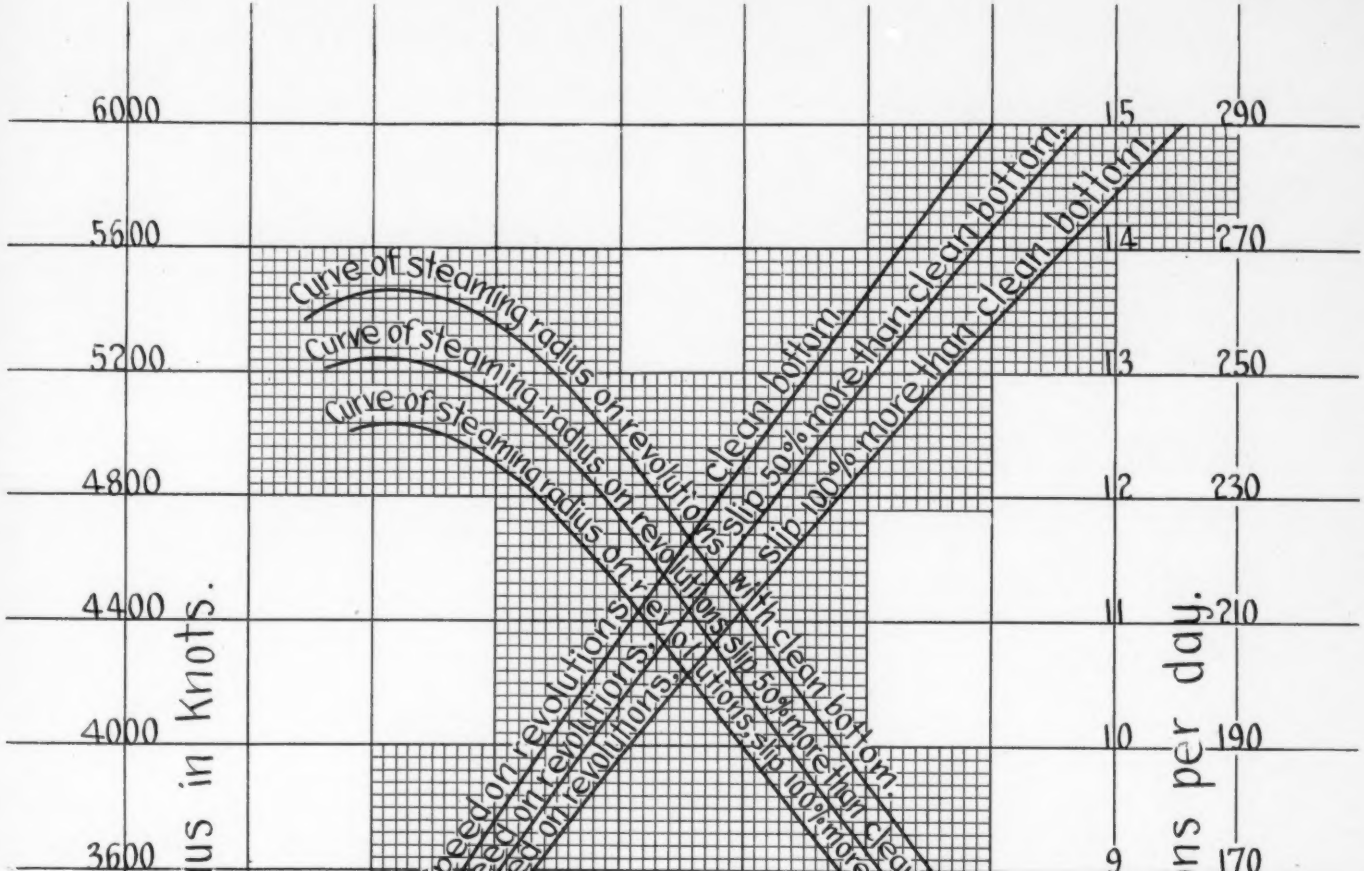
C.—SLIP 100 PER CENT. MORE THAN CLEAN BOTTOM.

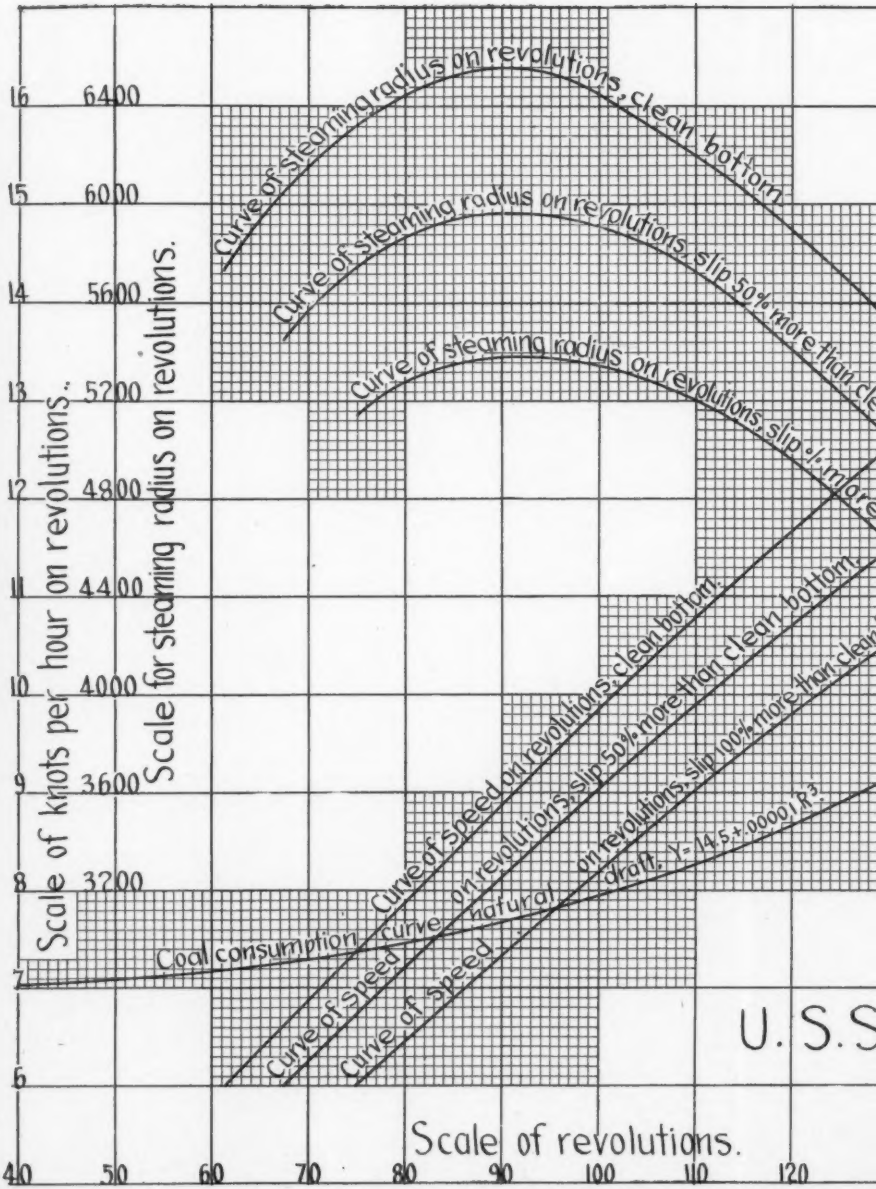
Speed, knots	...	...	6	7	8	9	10	11	12	13	14	15	16	17
Slip, per cent	...	...	30.7	29.7	29.2	29.1	29.6	30.8	32.4	34.4	36.4	...	...	...
Revolutions per minute	...	...	74.9	86.1	97.7	109.7	122.8	137.4	160.1	...	...	...	...	...
Total coal per day, tons	...	...	18.7	20.9	23.8	27.7	33	40.4	55.5	...	...	...	...	...
Steaming radius, knots	...	...	5,144	5,369	5,389	5,208	4,858	4,365	3,466	...	...	...	...	...

Most economical speed = 7.3 knots. Steaming radius = 5,390.

XU


U.S.S. Kearsarge.
Scale of steaming radius in miles.
3600
3200
2800
2400
2000
1600
Scale of revolutions.
40
50
60
70
80
90
100
Scale of Knots per hour.
2
3
4
5
6
7
8
9
10
Scale of coal consumption in tons per
170
150
130
110
90
70
50
30
Curve of coal consumption on revolutions, natural draft.
Speed on revolutions, natural draft.
Speed on revolutions, natural draft.
Speed on revolutions, natural draft.
Slip 100% more than clean bottom.
Slip 100% more than clean bottom.
Slip 100% more than clean bottom.
V=38.0013R^3


6000
5600
5200
4800
4400
4000
3600
Curve of steaming radius on revolutions, slip 50% more than clean bottom.
Curve of steaming radius on revolutions, slip 100% more than clean bottom.
Curve of steaming radius on revolutions, with clean bottom.
Clean bottom.
slip 50% more than clean bottom.
slip 100% more than clean bottom.
15
14
13
12
11
10
9
290
270
250
230
210
190
170
Miles run in knots.
Tons per day.


Curve of steaming radius on revolutions, clean bottom.
Curve of steaming radius on revolutions, slip 50% more than cle
Curve of steaming radius on revolutions, slip
Scale of knots per hour on revolutions.
Scale for steaming radius on revolutions.
16 6400
15 6000
14 5600
13 5200
12 4800
11 4400
10 4000
9 3600
8 3200
7
6
Coal consumption.
Curve of speed on revolutions, clean bottom.
Curve of speed on revolutions, slip 50% more than clean bottom.
Curve of speed on revolutions, slip 100% more than clean
Curve of speed natural draft, V=14.5+0.0001 d³.
U.S.S
Scale of revolutions.
40 50 60 70 80 90 100 110 120

105
95
85
75
65
55
45
35
25
15
5
Scale of coal consumption in knots on revolutions.
50% more than clean bottom
clean bottom
more than on clean bottom
more than clean bottom
J. S.S Cleveland.
120
130
140
150
160
170
180

A COMPARISON OF THE PERFORMANCES OF PROPELLERS U. S. S. *WYOMING.**

By Lieutenant J. P. Morton, U. S. Navy.

The steam engineering specifications for the U. S. S. *Wyoming* called for the manufacture of a spare set of propeller blades to be built after the vessel's trial and to conform to specifications that would be furnished by the Bureau.

These blades were designed with increased projected and expanded areas, increased diameter of propeller and decreased pitch, but after being in service the pitch was increased.

A comparison of the performances of the vessel with the original propellers and with the ones now in place, under the two conditions of pitch, may be of some interest as illustrating the great range of efficiency for a small change of pitch. The material point is not the improvement brought about by the change of blades, as that was generally anticipated, but in the startling effects produced by the change of pitch in the new blades. The new propellers were first installed with a pitch of 7 feet 6 inches. After about a year the pitch was changed to 8 feet 6½ inches. The following results will prove that this change was justifiable.

The original propellers were rights-and-lefts, three bladed, built-up, true screws, bent back five inches, and of manganese-bronze. The top tips of the blades turned inboard in the ahead motion. The pitch was adjustable from 7 feet 8.6 inches to 9 feet 4.4 inches.

The propellers now in use differed from the original in the particulars noted in the following table. Their pitch is adjustable from 7 feet 4 inches to 8 feet 6½ inches. The general appearance of the blades is shown in the sketches.

For the sake of brevity, the conditions enumerated in the following table will hereinafter be referred to as (1), (2) and

* For a description of the *Wyoming* see Vol. XV, page 83.

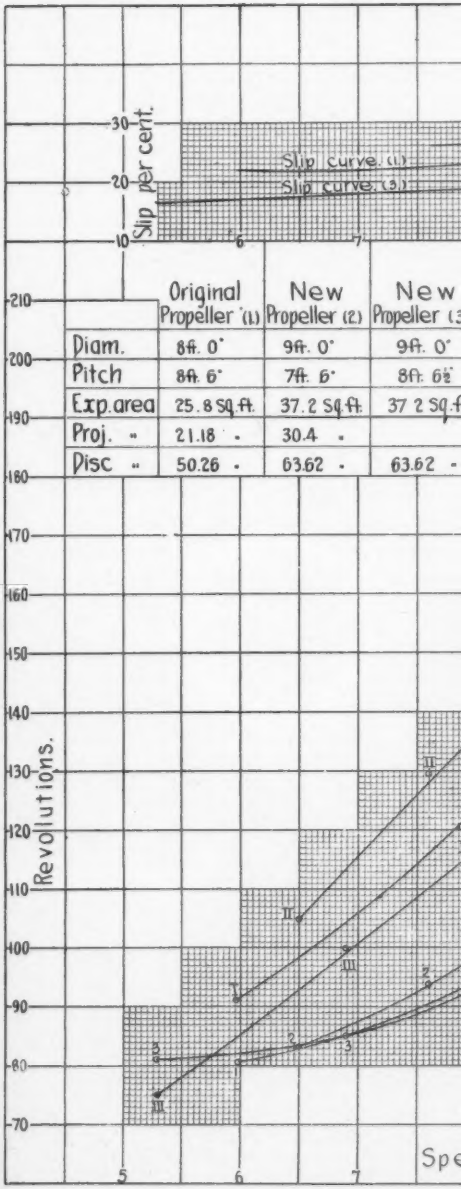
(3)—(1), original propellers; (2), new propellers as at first set; (3), new propellers as now set:

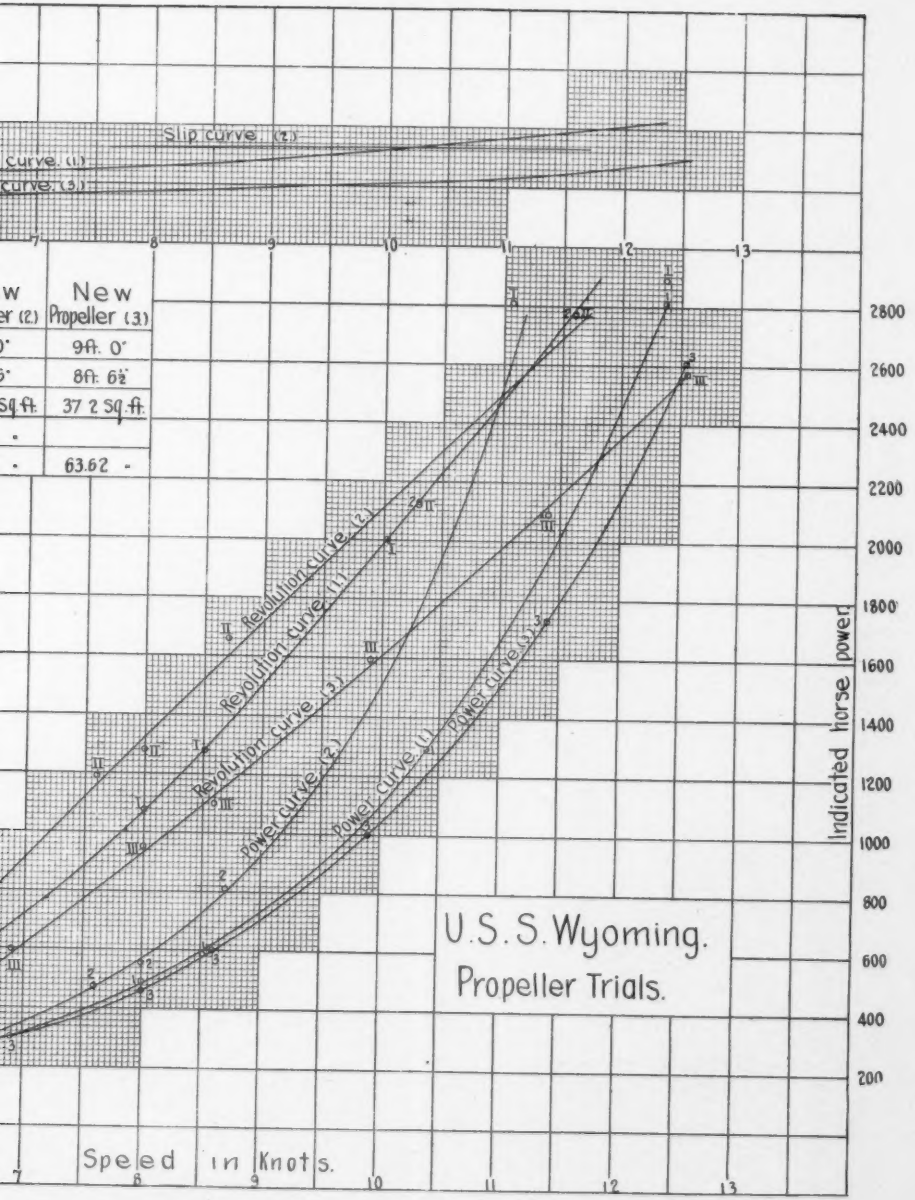
Condition.	(1)	(2)	(3)
Diameter of propeller, feet and inches	8- 0	9- 0	9- 0
Pitch, feet and inches	8- 6	7- 6	8- 6½
Expanded area, square feet	25.8	37.2	37.2
Projected area, square feet	21.18	32.4	...
Disc area, square feet	50.28	63.62	63.62
$\frac{P}{D}$	1.0614	.833	.949
Number of blades	3	3	3
Diameter of hub, inches	26.75	26.75	26.75
Length of hub, inches	23	23	23
Greatest width of blade, inches	39.5	50	50

The three trials, whose results will be touched upon, were made at intervals of about twelve months, and under similar conditions of hull and sea and wind, all trials being made just after the ship's bottom had been cleaned and in smooth water.

A comparison of the power curves shows no improvement as a consequence of installing the new propellers at the pitch of 7 feet 6 inches; on the contrary, a distinct loss is evident, greater power being required for all speeds except the lowest obtained. The points of similarity observed were: (a) excessive churning of the water in the wake, (b) high coal consumption, (c) relatively high speed of engines for moderate speed of ship; (d) inability to steer with one engine only in use. The great number of revolutions required for ordinary cruising speeds, and the consequent wear and tear in the engines, was especially objectionable and was the direct cause of the request to have the pitch increased. At a speed of 8.8 knots conditions below resembled a forced-draft trial; being in squadron most of the time this speed was required, and, as a result, the brasses were soon all in need of rebabiting, and additional bracings were required for the engine rockshafts, the old ones vibrating to an alarming extent. After a few days at this speed, when the tubes became a little dirty or the draft poor, the blowers were run to maintain the set speed. This was all done away with under conditions (3), and the marked improvement may be gauged from the statement that recently the

XL


Slip per cent.
30
20
10
Slip curve. (1.)
Slip curve. (3.)
6
7
Original Propeller (1)
New Propeller (2)
New Propeller (3
Diam. 8ft. 0" 9ft. 0" 9ft. 0"
Pitch 8ft. 6" 7ft. 6" 8ft. 6½"
Exp. area 25.8 Sq.ft. 37.2 Sq.ft. 37.2 Sq.f
Proj. " 21.18 " 30.4 "
Disc " 50.26 " 63.62 " 63.62 "
Revolutions.
210
200
190
180
170
160
150
140
130
120
110
100
90
80
70
5
6
7
Spe


Slip curve. (2)
curve. (1.)
curve. (3.)
w
er (2)
New
Propeller (3)
9ft. 0"
8ft. 6½"
37 2 Sq ft.
63.62 -
Revolution curve. (2.)
Revolution curve. (1.)
Revolution curve. (3.)
power curve. (2.)
Power curve. (1.)
Power curve. (3.)
U.S.S. Wyoming.
Propeller Trials.
Indicated horse power.
2800
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
Speed in Knots.
7
8
9
10
11
12
13

trip from Acapulco to Panama, a distance of about 1,540 miles, was made at 8.8 knots with three, instead of four, boilers, and without the need of blowers. The same trip with the old pitch was made at 8.7 with four boilers, blowers going most of the time, and with the expenditure of 31 tons of coal daily, as compared with a consumption of 24 tons on the last trip. The coal used was from the same mine, and other conditions were practically identical.

One of the most remarkable effects of the change in pitch is that, for the first time in the ship's history, she can be steered perfectly with one engine turning over. Under conditions (1) and (2) the ship could not be kept on a steady course with one propeller, but at once began turning against the helm; under condition (3) she can be kept steady in any kind of a sea, with from 15 to 25 degrees of helm. When, during a gale on July 18, 1905, the starboard shaft broke, the ship ran on her course for twenty-seven hours and came safely into port.

The increase in economy may be inferred from the speed, slip and power curves, and also saving to bearings and engine framing and bracing, due to slower speed of engine, the necessary repairs after a long trip being very considerably reduced.

A summary of the advantages possessed by the new propellers at the new pitch follows:

(a) Reduction in coal consumption; *e. g.*, at the cruising speed of 8.8 knots, as mentioned above.

(b) The saving due to the use of three boilers instead of four at low speeds.

(c) Reduction of revolutions and corresponding lessening of wear due to vibration and the racking strains in the engines.

(d) Increase of maneuvering qualities by reason of improved steering, and the further practicability of steering with only one engine in use.

By adopting the reasoning employed by Assistant Engineer R. E. Carney, U. S. N. (Retired), as given in his article printed in the JOURNAL, volume XV, page 823, we have the following table for the three conditions, using the notation employed by him:

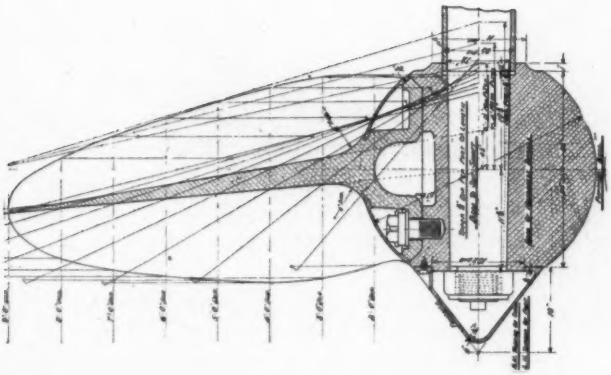


Fig. 2.—New Propeller.

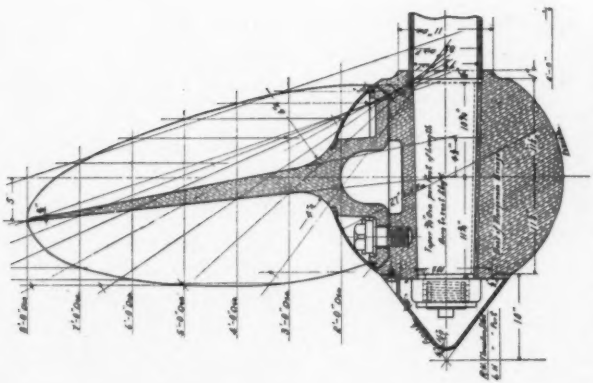


Fig. 1.—Original Propeller.

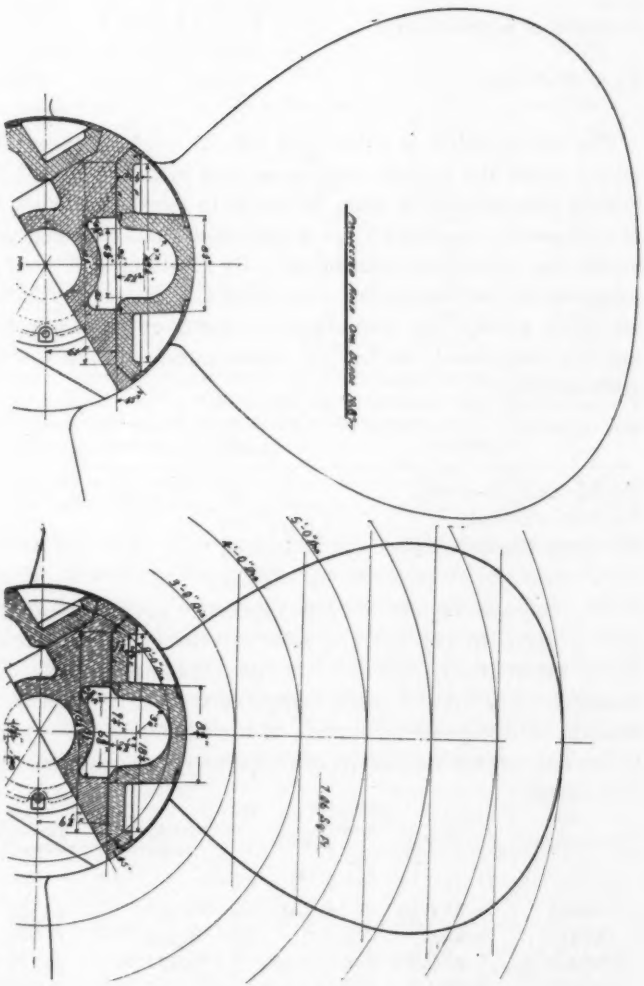


Fig. 2a.—New Propeller.

Fig. 1a.—Original Propeller.

Condition.	(1)	(2)	(3)
CA, ...	13.189	12.826	14.003
Advance of screw (ship), . .	6.0141	5.632	6.401
Reverse current, ...	7.1749	7.194	7.602

The above table is calculated for the highest speeds obtained under the various conditions, and in this, as in all following comparisons, it must be borne in mind that with the new propeller, condition (3), a higher speed was obtained than under the other two conditions. By resolving CA into its components and comparing them with CN we get the following table, giving the percentages of power employed in driving the ship ahead, driving the water astern, and in wastage athwartships:

Condition.	(1)	(2)	(3)
To advance of screw, . . .	35	38	37
Reverse current, ...	42	48	45
Wastage athwartships, . .	23	14	18

The increase of efficiency is especially noticeable in comparing (3) with (1), but the relatively low speed obtainable under (2) probably accounts for the rather unfavorable light in which (3) is viewed when compared with (2). This would seem to be dissipated by a study of the following table, which is worked out for various speeds, especially the common speed of 8 knots.

Revolutions.	H.P.	Speed. (1)	Advance of screw.	Reversed current.
91.5	211.55	5.97	6.612	6.557
124	495	8	6.344	6.825
134.2	589.99	8.57	6.4715	6.7175
170.5	1,295.5	10.4	6.1813	7.0077
210.3	.....	11.08	5.9403	7.2487
214.7	2,811.01	12.37	6.0141	7.1741

Revolutions.	H.P.	Speed.	Advance of screw.	Reversed current.
		(2)		
104.9	265	6.5	5.844	6.982
129.5	475	7.6	5.947	6.879
134	557	8	6.04	6.785
153	810	8.7	5.826	7
176.3	2,120	10.3	5.927	6.899
208.7	2,770	11.6	5.632	7.194
		(3)		
75	220	5.29	8.26	5.743
100	300	6.93	7.022	6.981
117.5	460	8	6.899	7.104
125	600	8.61	6.977	7.026
149.7	1,000	9.9	6.688	7.215
174.5	1,730	11.4	6.62	7.383
198.5	2,610	12.54	6.401	7.602

NOTE.—The above results were obtained in trials over the measured mile. During trials (1) and (2) efforts were made to steady ship on a course with one engine stopped, but it was found to be impracticable. During (3) trial the following results were obtained :

	Revs.	Speed.	H.P.
Running starboard engine only	173.5	8.72	1,107.16
Running port engine only	184	8.18	1,518.06

The following data was obtained under condition (3) on August 28 and 29, 1904, during a sea run, under most favorable conditions. The coal was carefully weighed. The patent log was checked by later runs over the measured mile. The steaming radius was calculated from the coal consumption of the short runs which were of about four hours duration at each speed. The ship's bottom had recently been cleaned; the sea was smooth, and wind very light and from ahead.

Revs.	Knots.	Coal per day.	Steaming radius.	Slip.	H.P.
		Tons.	*Knots.*		
100	7.07	18.75	3,298	16.1	334.34
110	7.65	20.35	3,107	17.4	418.21
120	8.45	22.35	3,380	16.4	470.8
130	9.3	24.6	3,303	14.8	654.65
140	9.9	27.2	3,183	15.8	831.91
150	10.7	29.7	3,158	15.4	959.99
160	11.1	32.5	3,198	17.6	1,138.9

TESTS OF THE CAVITE STEEL FLOATING DRY DOCK.

By Civil Engineer A. C. Cunningham, U. S. Navy, Member.

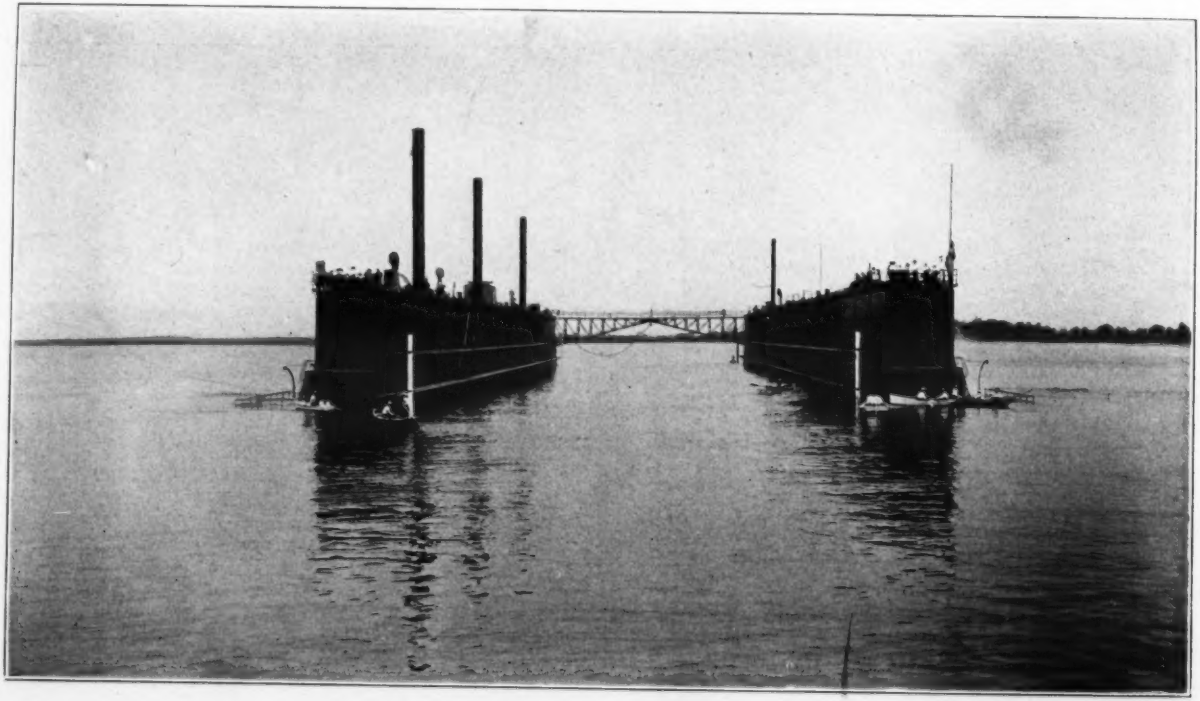
In the early part of June, 1905, the Maryland Steel Co. reported the Cavite self-docking steel floating dry dock ready for test, having completed the same in one month's less time than the twenty-seven allowed by the contract.

As in the case of the Government floating dock at New Orleans, built by the same company, the Cavite dock was constructed in a shallow basin, and on completion the basin was flooded and the cofferdam at the entrance removed. While flooding the basin the valves of the dock were left open and the dock allowed to fill, so that it continued to rest on the pile and timber foundation on which it was built until the time selected for floating and towing it to the mouth of the Patuxent River, the place selected for test.

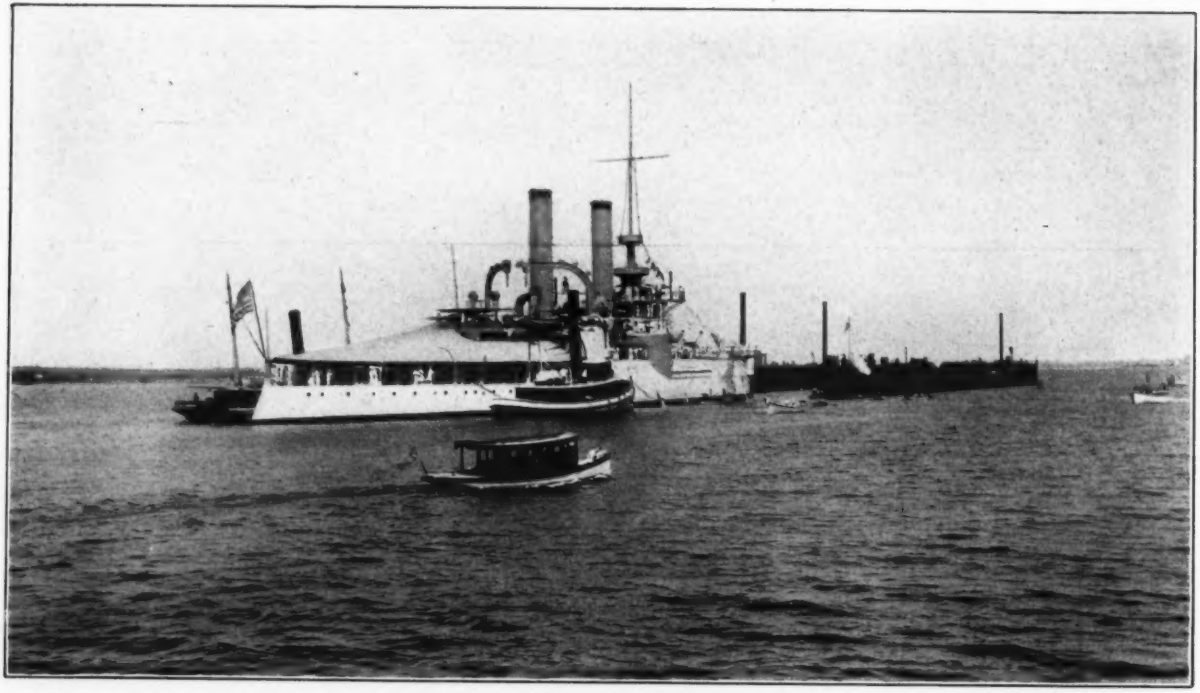
When the dock rose from its timber bed it was christened the *Dewey*, after the Admiral of the Navy, by Miss Grace Endicott, the daughter of Rear Admiral M. T. Endicott, Chief of the Bureau of Yards and Docks, Navy Department, under whose administration the dock had been built, and who represented the Navy Department when the dock was floated.

In the preliminary test of the New Orleans floating dock a collier of 6,000 tons displacement was used. The increased confidence in this class of dock for warships, and in the *Dewey* dock in particular, is shown by the fact that one of the latest and heaviest ships, the armored cruiser *Colorado*, was selected for the preliminary test.

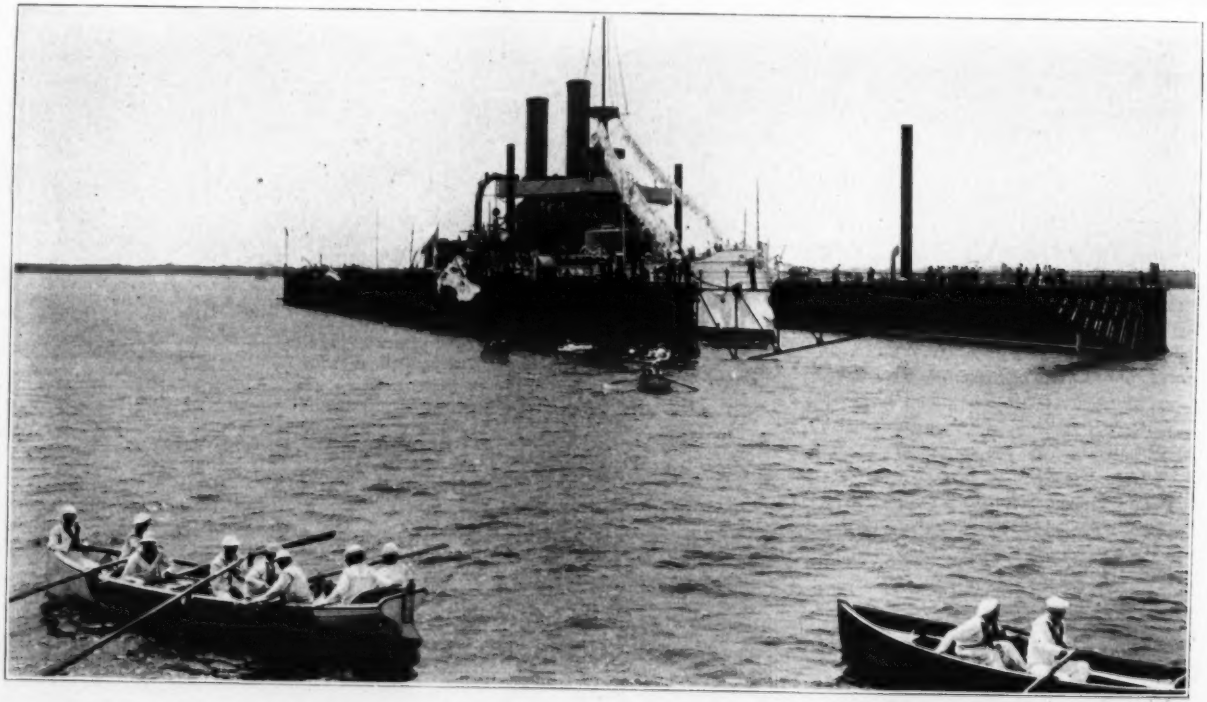
Before docking the *Colorado*, the dock was sunk to twenty-nine feet draught over the keel blocks and then pumped up light to a freeboard of two and one-half feet. The sinking was



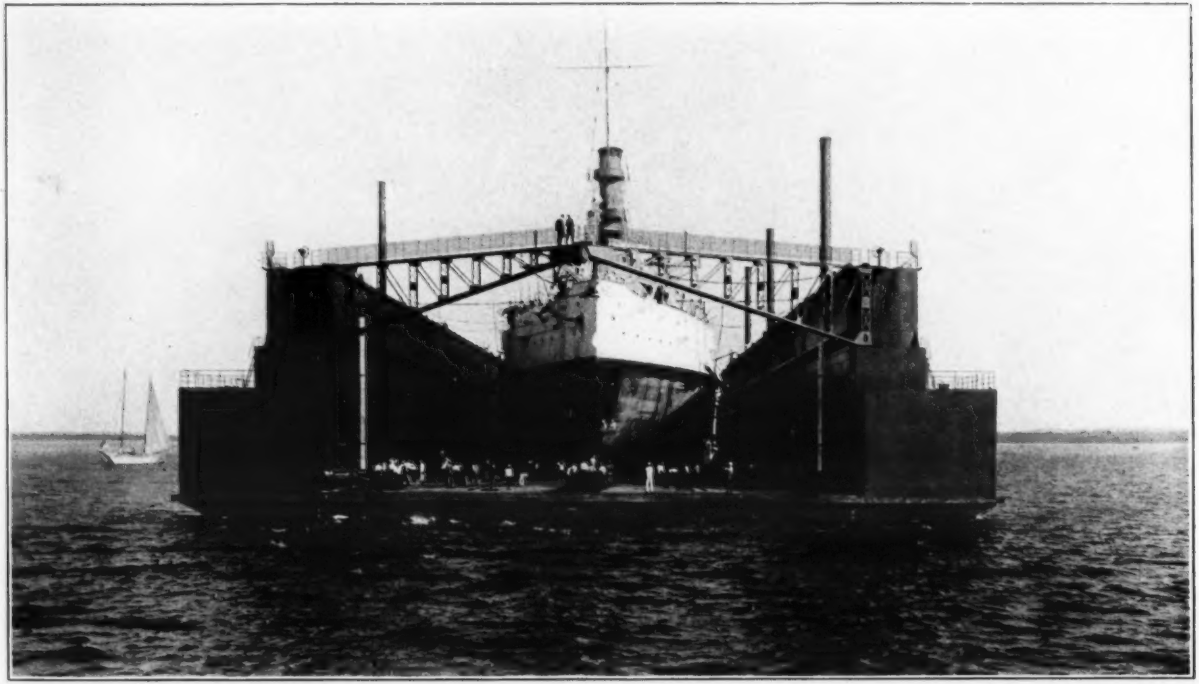
U. S. Floating Dry Dock "Dewey."—Dock Sunk Ready to Receive Ship.



U. S. Floating Dry Dock "Dewey."—U. S. Battleship "Iowa" Entering Dock.



U. S. Floating Dry Dock "Dewey."—U. S. Battleship "Iowa" Centered and Ready for Lifting.



U. S. Floating Dry Dock "Dewey."—U. S. Battleship "Iowa" Completely Docked.

accomplished in one hour and thirty-six minutes and the pumping up in one hour and two minutes, showing that the dock is of very rapid action. On this test it was also demonstrated that little or no trimming was required, the dock sinking and rising practically level with all valves open.

The U. S. S. *Colorado* was docked on June 23, 1905, having a displacement of 13,300 tons at that time. The main and docking keel blocks were all set at the same height. In this preliminary test no effort was made to secure speed, and one-half hour was used in making flushing and fire connections. The elapsed time from when the ship landed on the blocks until the keel came out of water was two hours and sixteen minutes. Pumping was continued until the dock had a uniform freeboard of two and one-half feet, only enough excess of water being retained in the side walls and end compartments to give the necessary trim. The *Colorado* was carried on the dock about twenty-four hours without changing the water ballast. When the dock had reached a freeboard of two and one-half feet with the *Colorado*, the deflection on the main keel line in the five hundred feet of length of the dock was about one-quarter of an inch; after about twenty-four hours the deflection in five hundred feet increased to about one and one-sixteenth inches. After undocking the *Colorado* the dock was found to have practically straightened without retaining any set.

After the undocking of the *Colorado*, deflection observations were continued for three days, and variations in deflections with the dock unloaded, due to temperature changes, of seven-eighths of an inch were noted.

The battleship *Iowa* was docked on June 27, 1905, for a record test, having a displacement of 11,600 tons at the time, and was carried on the dock for forty-eight hours. The specifications required that a 16,000-ton ship should be raised in four hours from the time the ship took the blocks until the keel was out of water. For the equivalent of a 16,000-ton ship the dock was pumped to a freeboard of four and one-half feet. From the time the *Iowa* took the blocks until the keel

was out of water was one hour and thirty-seven minutes; to the time the dock had a freeboard of four and one-half feet, two hours and forty-two minutes. During the docking of the *Iowa* one of the three pumping engines was out of commission for forty-two minutes with a slipped eccentric, so that the actual time of operation of the dock is about half that allowed by the specification.

The *Iowa* was docked by uniform pumping, as in the case of the *Colorado*, and carried for forty-eight hours without change of water ballast in the dock. The specification required that when a ship had been docked by uniform pumping until the dock had a freeboard of two feet the deflection in the entire 500 feet of length of the dock should not exceed three inches. When the dock reached a freeboard of four and a-half feet with the *Iowa*, the deflection was about two inches. During the first twenty-four hours, the dock remaining uniformly pumped, the deflection increased to four inches in the 500 feet, and during the second twenty-four hours showed a recovery to three and three-eighths inches.

Immediately following the undocking of the *Iowa* the dock was pumped up to the same depth of water in the compartments as when the ship was docked, which gave a freeboard of nine feet and six inches, and it was found that the dock had a hog of one inch. During the night this hog disappeared, and early the next morning was a half inch sag. The greatest deflection in the bearing length of the *Iowa* while carried on the dock was about one and three-quarter inches. The deflection observations indicate that there was no permanent set caused by the docking, and that temperature variations may cause considerable hog or sag.

After the undocking of the *Colorado* the main and docking keel blocks were found to be uniformly indented about one-sixteenth of an inch with no crushing. No change was made in the blocks for the *Iowa*, and after undocking she was found to have rested even more easily than the *Colorado*.

With two feet of freeboard and one foot of water remaining in the pontoons the carrying capacity of the dock is 18,500

tons, and a 20,000-ton ship could readily be docked with sufficient freeboard to admit of easily working on its bottom.

The results of the docking tests with the *Dewey* dock show that it is considerably in excess of the contract requirements in strength, time of operation and capacity, and is in all respects a very noticeable advance on all floating dry docks which have been so far projected or built. The greatest innovation is the requirement of uniform pumping. This not only insures safety from careless or unskilled handling, but makes it possible to dock nearly all ships with keel straight, or with as much hog or sag as circumstances may render desirable, by suitably distributing the water ballast in the dock. The ship may be also hogged or sagged while on the dock, should occasion arise for so doing. With thirty feet of water over four-foot keel blocks the side walls of the dock have a freeboard of eleven feet. By taking the keel blocks down to two feet and sinking until the side walls have a freeboard of three feet, forty feet of water may be had over the blocks, so that a ship may be taken into the dock in any condition of disablement.

Although a heavier ship, the deflections with the *Colorado* were less than with the *Iowa*, on account of the much longer bearing length of the *Colorado's* keel. With the uniform pumping the strains caused by the *Iowa* were much greater than will be caused by any other ship in the service, projected or built; so that the dock has had the most severe test that can be imposed on it.

The dock was constructed under the supervision of Civil Engineer Leonard M. Cox, U. S. N., and tested by a Board of naval officers, consisting of Captain Adolph Marix, senior member; Naval Constructor J. H. Linnard, Commander J. F. Parker, Commander W. F. Worthington, Naval Constructor J. H. Rock, Civil Engineer A. C. Cunningham and Assistant Civil Engineer J. S. Shultz. The Board was assisted in the tests by Civil Engineer Cox.

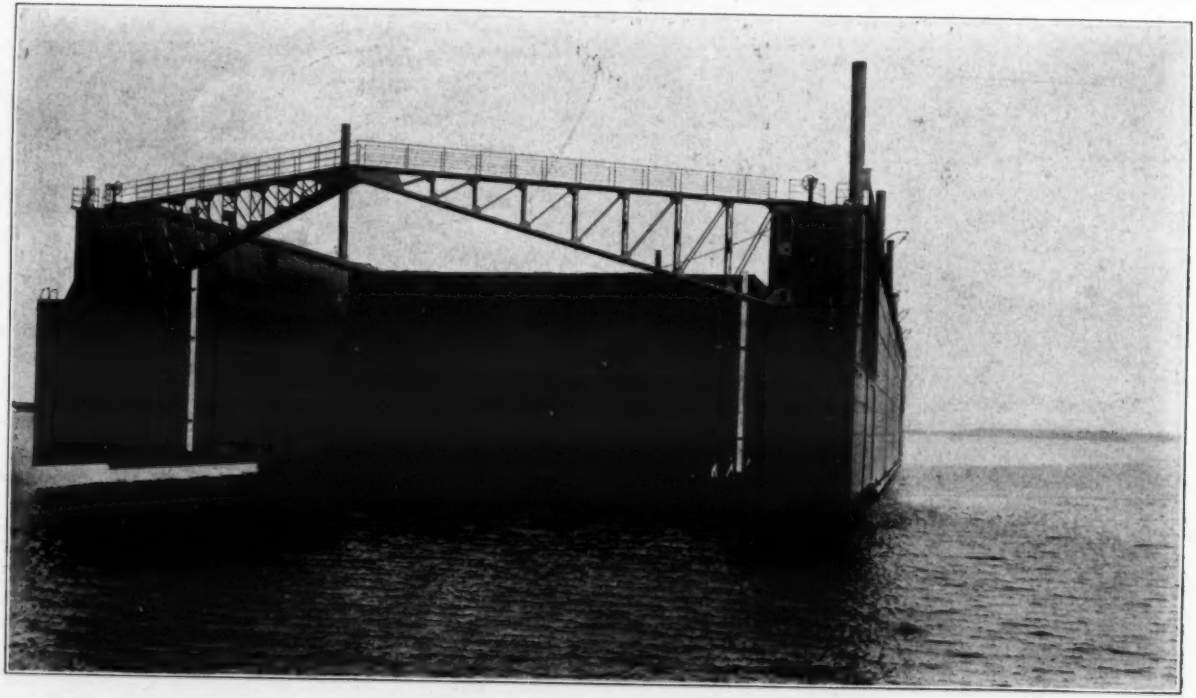
THE SELF-DOCKING TESTS OF THE STEEL FLOATING DRY DOCK FOR CAVITE, P. I.

By Civil Engineer Leonard M. Cox, U. S. Navy.

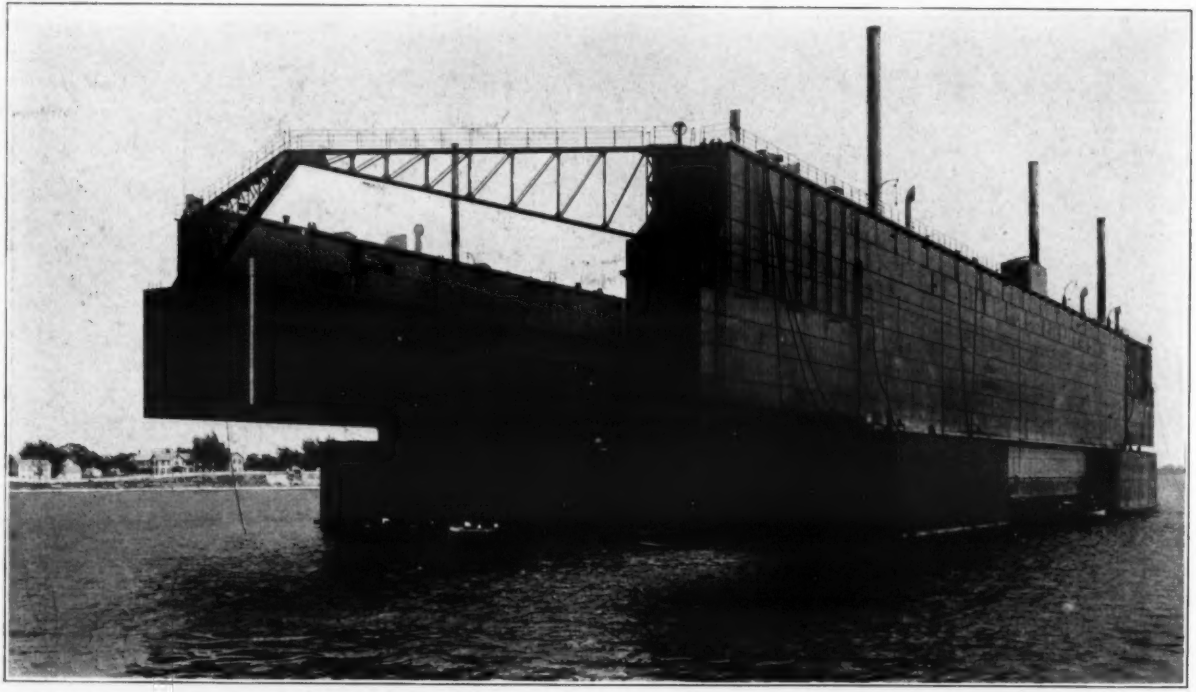
In the specifications forming a part of the contract between the Government and the Maryland Steel Company for the construction of the steel floating dry dock for Cavite, P. I., it is provided that the structure, after completion, shall be given a thorough working test before final acceptance. This test, as provided for in the contract and its specifications, comprises three distinct operations: The lifting, within a given time, to a specified freeboard and with limited deflections, of the heaviest type of cruiser; the lifting, under similarly specified conditions, of the heaviest battleship; and, finally, the lifting of its own hull clear of water.

On June 10, 1905, the dock was launched from the building basin at the manufacturers' works and was immediately towed to the mouth of the Patuxent River, the site selected for the official tests. After preliminary trials of the machinery, the dock was declared ready for testing on June 17, 1905, just one month before the expiration of the contract time of twenty-seven months from the date of award. On June 23d, the cruiser *Colorado* was docked, followed by the battleship *Iowa* on June 27th, and preparations were at once begun for the self-docking trials.

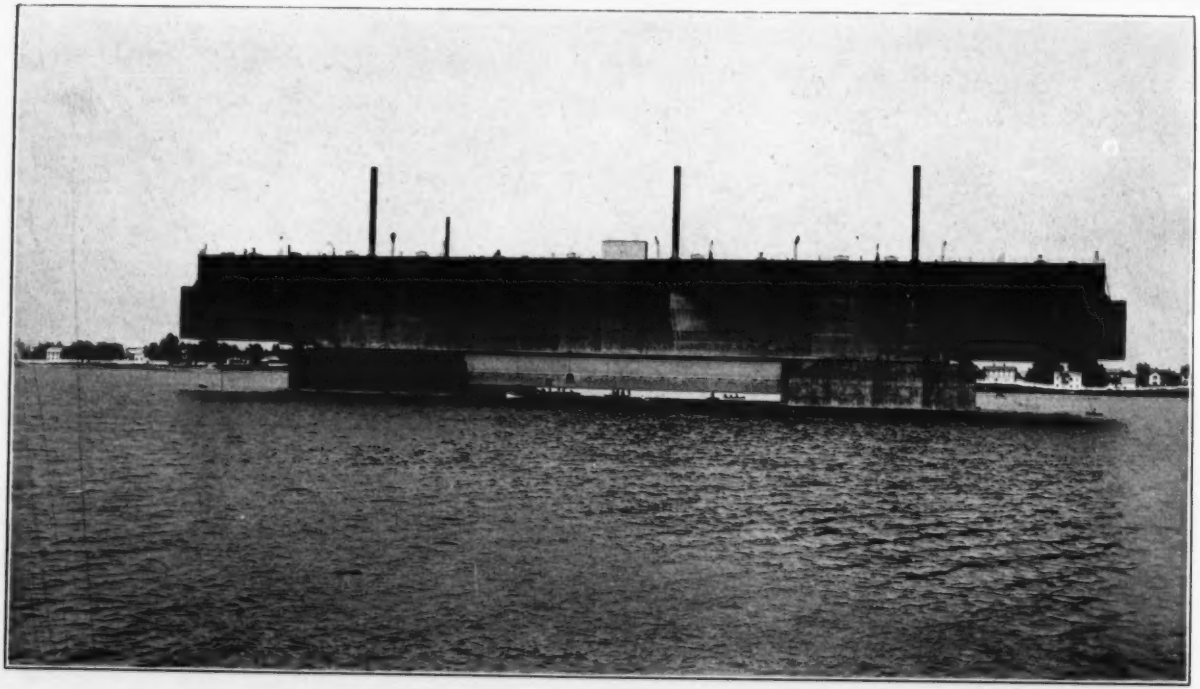
An article appearing in this issue of the Journal (page 714) fully describes the successful docking of the *Colorado* and the *Iowa*. This article, coming as it does from the officer who wrote the specifications under which the dock was constructed and who was closely connected with its preliminary design, is of peculiar interest, but, as it covers only the ship-docking tests, it may not be out of place to add a brief account of the subsequent self-docking operations.



U. S. Floating Dry Dock "Dewey."—End Pontoons Self-Docked on Center Pontoon.

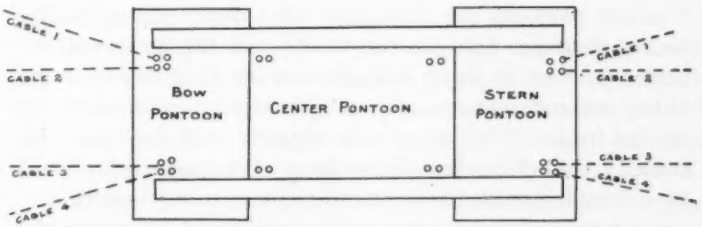


U. S. Floating Dry Dock "Dewey."—Center Pontoon Self-Docked on End Pontoons.



U. S. FLOATING DRY DOCK "DEWEY."—CENTER PONTOON SELF-DOCKED ON END PONTOONS.

In regard to the desired self-docking qualities of the dock, the specifications provide that it "shall be so designed and arranged as to be readily self-docked without the aid of divers or auxiliary constructions;" that the working stress of any portion of the dock or its connections shall be limited to "15,000 pounds per square inch in self-docking, with a wind pressure of 30 pounds per square foot of exposed surface;" that "when self-docked all under-water portions shall be raised to a height of not less than 5 feet and shall be safely and readily accessible for inspection, painting and repairs;" and, that with the dock at light-draught line, "all self-docking and strain-transmitting connections shall be above water."



Although plans and complete descriptions of the structure have already appeared in the JOURNAL (Vol. XV, page 472) and other technical publications, it may help to a clearer understanding of the operations to give, briefly, a general outline of the design in so far as it affects the self-docking features before attempting to describe the working tests. The dock is built in three sections. The center section, or pontoon, is 316 feet in length, with side walls overhanging 80 feet on either end. Two end pontoons, each 90 feet in length, with low independent side walls, are attached to the center pontoon in such a way that the overhanging side walls of the latter are enclosed between the independent walls of the end pontoons and rest directly on their decks when the sections are connected for docking ships. There are vertical and horizontal connections between the pontoons, the vertical connections consisting of seven elements, each made up of forty-four 2-inch

bolts; the horizontal, of four elements, one along each edge of the overhanging side walls, each made up of ninety-six 1½-inch bolts. The lower ends of the vertical connections are out of water at light draught.

The self-docking is accomplished in two stages. After removing the connection bolts, the end pontoons are hauled clear of the overhanging walls of the center pontoon, turned so that the direction of their lengths is perpendicular to the axis of the dock, and brought over the deck of the center pontoon and centered on blocking, as in the docking of ships. The center pontoon is then pumped to the desired freeboard by the main pumping engines. The second stage of the operations has for its object the lifting of the center pontoon. The end pontoons are submerged to a depth giving twelve inches clearance between the blocks and the bottom of the center pontoon at light draught, and are then drawn under either end and centered, after which they are pumped to the desired freeboard by means of a separate pumping plant installed in the independent side walls, to which steam is furnished by a flexible hose led from the forward and after boilers.

On Saturday, July 15th, 1905, the work of arranging the blocks of the center pontoon for receiving the two ends was finished, and all vertical connection bolts, together with half of the horizontal bolts, were removed. On Monday, cables 1 and 2 (see accompanying sketch) were slipped from the end pontoon bitts, starboard side, forward, and made fast by hawsers to bitts on the top of the overhanging side walls. On Tuesday, cables 3 and 4, forward, were similarly slipped and belayed on the port side-wall bitts. All horizontal connections were removed and steel drift pins placed in the bolt holes at intervals of about twenty-five feet, with men standing by for each pin. At 12·15 P. M. the flood valves of the forward pontoon were opened, and ten minutes later the horizontal connection diaphragms came gently apart. With the aid of a small tug boat, the end pontoon was hauled clear of the overhanging side walls at 12·53 P. M., and made fast against the starboard side of the center pontoon at 1·35 P. M. Cables 2 and 3 were

then slipped from the tower-deck bitts and made fast to the forward-deck bitts of the center pontoon with two shots of chain which had previously been detached from cables 1 and 4. An anchor-hoy and tug was used in shifting moorings and handling cables.

On July 19th, cables were slipped from the after pontoon bitts and made fast to the side-wall bitts as was done in the case of the bow pontoon. All horizontal connection bolts were removed and the flood valves opened at 10·20 A. M. The pontoon began to drift out with the tide at 10·49, and was hauled to the port side of the dock and made fast at 12 M. Cables 2 and 3 were then slipped from the tower bitts and made fast to the after center pontoon bitts as for the bow pontoon.

On July 20th, the center pontoon was submerged to an average draught of thirty-three feet, giving a little over twelve inches clearance between the bottoms of end pontoons and the blocking for a draught of nine feet. The end pontoons were then hauled in and centered over the keel blocks. Pumping was commenced at 12·26 P. M., the blocks were awash at 1·25, and the pumps were stopped at 1·40 with a freeboard of eighteen inches, the total time of pumping being one hour and twenty minutes. All mud valves in end pontoons were opened at 1·00 P. M., and the compartments drained with the falling water.

During the next three days the pontoons were held on the blocks and the time was spent in cleaning and repainting such parts of their bottoms as seemed to require it, and in arranging the blocking on the ends for seating the center pontoon. On July 24th the mud valves of the end pontoons were closed at 8·40 A. M., the flood valves were opened at 9·15 and closed with the pontoons afloat at 10·00 A. M. The pontoons were lashed together in tandem and drawn out with the tide, being clear of the dock at 11·31 A. M. Both pontoons were then made fast to the starboard side of the center pontoon. When floated, the end pontoons had an average draught of 4.59 feet, making their joint displacement 3,164 tons.

The 25th of July was consumed in arranging the blocking, and on the 26th the work of docking the center pontoon was begun. Cables 2 and 3 were slipped and led by hawsers to side-wall bitts, the dock being held in the meanwhile by cables 1 and 4. At 8·30 A. M. the stern pontoon was sunk to sufficient depth to permit its being hauled under the overhangs, and at 11·40 it was in position and ready for sinking to full draught. At 1·05 the flood valves were opened, and at 3·50 P. M. the pontoon had reached an average draught of thirty-one feet, the sinking being carried on very deliberately on account of the possible effect of the strong ebb tide, nearly always to be contended with at this point. After reaching this depth the pontoon was drawn under the center pontoon and centered over the blocks. The pumps were started and continued until the pontoon reached a draught of twenty-eight feet, and the center was firmly seated.

On July 27th the dock was simply held in position and observations made for possible settlement. On July 28th the bow pontoon was drawn into position, sunk to a draught of thirty-one feet, drawn under the center pontoon and, at 2·20 P. M., pumped to a uniform draught with the stern pontoon. Both end pontoons began pumping full speed at 2·20 P. M. and reached a freeboard of eighteen inches at 7 P. M. After holding the dock in this position for four days, the work of undocking was commenced, but, on account of bad weather, this work was discontinued and the end pontoons were held at twenty-six feet draught until August 2d, when the stern pontoon was hauled from under the center and connected in its proper position. Two more days were lost on account of the weather, and the bow pontoon was withdrawn and connected on August 4th. No difficulty was encountered in entering the connection bolts, and each pontoon was sunk, hauled out, pumped to proper draught and connected in twelve hours time.

The total working time consumed in the self-docking tests, exclusive of the time spent on the blocks, is about fifteen days, and, when it is taken into consideration that this work is

absolutely new, that it was necessary to proceed with caution during the first test, and that the crew was naturally unused to its work, it would seem probable that the time for future self-dockings could easily be reduced to ten or eleven working days.

Numerous devices have been exploited in the past, all claiming more or less novel means to enable floating docks to lift their own hulls for inspection and repairs. Few of these devices appear practical, and still fewer have been actually tried on large structures. The most prominent among the self-docking types is the Clark & Stanfield modification of the Rennie type, represented in this country by the naval docks at Algiers, La., and at Pensacola, Fla. The Algiers dock was successfully self-docked by her builders during the acceptance tests, and required for the operation about forty days. It succeeded in lifting the pontoons to a height of four feet and the side walls to a height of twenty-one inches above water. The Pensacola dock has docked her intermediate pontoons, but is only now, for the first time, attempting to lift the end sections. So far as I am aware, the new Bermuda dock has never been completely docked, nor has the new naval dock at Pola, Austria, the latter being an entirely new design.

The Cavite dock has, so far, proven a complete success, and the self-docking features are particularly satisfactory. In the light of experience with other self-docking types, it would seem that, as regards the saving effected in time and labor, facility of control and simplicity of operation, the self-docking problem has been satisfactorily solved and the greatest objection to the floating dock *per se* completely eliminated.

The self-docking tests were conducted by a Board of which Captain Adolph Marix was senior member, and Naval Constructor W. G. Du Bose and Civil Engineer Leonard M. Cox, members.

SUGGESTIONS AS TO THE ADAPTABILITY OF A UNIVERSAL STANDARD FLANGE LIST FOR MARINE WORK.

By Luther D. Lovekin, Chief Engineer of the New York Shipbuilding Co., of Camden, N. J., Associate Member.

The subject of flanges for all purposes where pipes are required to be connected by such means is one of the most important details in connection with modern engineering, and one which, without doubt, has received the least attention among designing engineers. Too often does it occur in the drawing rooms of large establishments that when a flange list is to be made out it is given to some young draftsman of very limited experience, with a view, no doubt, of economizing, and when finished we find this new list embodies the mistakes of the past as well as some new features which are liable to add to the already innumerable troubles existing in this line.

With some knowledge of the troubles existing in this line (due to various causes), the writer decided that the time was very close at hand when engineers must give this subject their very best attention, and this will be very apparent to all when we consider for a moment the great tendencies toward the use of higher boiler pressures and their consequent economy. As a result of this, during the year 1899, I took my initiative step in this line, in first inventing a new form of pipe joint, and in order to be certain that I would cover all possible merits in connection with the same, I went at the designing of this flange with as much pleasure as though I were designing an engine. I made all sorts of calculations, such as would apply to this line of work, with a view of placing the metal of the flange in the place where it was most needed, and before I had

finished my first series of designs and had gathered together all the data I could, I almost concluded that this branch of engineering work had been left to the old and familiar method of "rule of thumb." Be it as it may, I cannot refrain from expressing the view that, while prominent engineers have frequently taken this subject up, they have seldom given the matter serious thought, but, instead, as a rule, when a matter of this kind has been brought to their attention, they have simply asked what the trouble was, and in the case of a flange springing between bolts, or the bolts themselves being weak, they have either said to the draftsman "Put another $\frac{1}{8}$-inch on the thickness," or "Put the next size larger bolts in, and I think you will avoid the trouble."

How many engineers have ever taken the trouble to analyze the forces set up in flanges of the various sizes with a view of determining what would be the most suitable shape of a flange, the amount of surface a gasket should have in order to produce a good joint, and, furthermore, to see that whatever flange was used would, as far as practicable, be of the least weight consistent with strength? It seems to me there has been more talk on such items as *multiples of four bolts*, *being absolutely essential to any standard*, and the thoughts as to how this or that suggestion would affect the valve maker, with his *numerous patterns* on hand, than as to how this whole matter was going to affect the engineer whether at sea or on shore, as well as the owners of our large vessels and industrial works.

There is no doubt in my mind but that the industrial establishments throughout our whole country are suffering from leaky joints in steam pipes, and spending thousands of dollars on various kinds of gaskets or other jointing material in order to remedy defects which are not always due to the kind of material used for such gaskets, but are chiefly due to a lack of strength in the flanges themselves. The writer firmly believes in standards, and appreciates the fact that others before him have made standards, but appreciates the additional fact that no good standard flange list appears to be in general use at the present time for the *higher pressures;* in fact, some of our

best concerns in New York City today are ordering forged-steel flanges of the same dimensions as the standard cast-iron ones in use at the present time for large power plants. Surely they are not doing this unnecessarily? No, for I have been informed by parties well versed in this line that they are ordering these ponderous steel flanges for safety only, and for this latter cause we cannot help but admire them.

However, we all know that the weakest link in a chain will be the one to first break under great strain, and, therefore, why not adopt some logical reasoning with this subject, just as we do in all other engineering subjects, and then when we have finished our work we can stand on some firm foundation, particularly so after a complete series of tests are made which confirms the theory of our work.

Personally, I see no reason for any variation in the practice of pipe fitting of any character; what we all want is a universal standard, but we will never have this so long as the mechanical engineers of the world adhere to the *multiple of four bolts*, for we all know that with ship work, where every pound of metal wasted means just so many pounds less cargo carried, and, naturally, so much less earning capacity of the vessel. Furthermore, in our naval work, we are compelled to pay a penalty of $500.00 per ton, over and above the weights specified by the Engineering Department.

With such restrictions as these resting upon the designers, builders or owners, it becomes of interest to all to see that there is no unnecessary excess of weight carried on ship-board, such as would necessarily follow were we to attempt the use of the bulky and yet weak flange list known as the *Mechanical Engineers' Standard*, which, in most cases under my supervision, has proven so inefficient in the number of bolts used as to often require the placing of additional bolts between these *multiples of four* after the work has been erected. Personally, I do not believe there is a shipbuilder or marine engineer in the United States who would support such a standard as is used in the general commercial work for pressures above 150 pounds. If this is correct, why, then, should we not all

unite in one standard, at least for the higher pressures, say from 150 pounds steam pressure and upward? This would enable us all to use valves or fittings of any character for the conveyance of steam, which we all know should be accomplished in the same manner, whether it be aboard ship or in our power houses. There is positively no good reason why we should not have that same factor of safety in our power houses that we have in our ships, and *vice-versa*.

Furthermore, in my opinion, any standard flange list should have the joints or flanges *equal at least to the bursting strength of the pipe itself*, so that we could rest assured that, in the case of water hammers and other shocks of a similar nature we had at least done our best from an engineering standpoint.

The writer having given his best attention to the subject of pipe flanges for several years, and constantly observing the various troubles that were brought to his attention, begs to submit to the marine-engineering world a series of flanges which have been designed very carefully, with a view of overcoming all possible objection, and will be very glad to hear from any of our engineers relative to the same. It might be well to state that my reason for calling particular attention to the marine-engineering world in this direction is due to the lack of any one standard which covers both the Navy Department and the merchant marine.

With this in view, I present herewith a full list of flanges (see sheets Nos. 1, 2 and 3), from 2 inches to 20 inches diameter, designed for 300 pounds working pressure, the same diameter of flange, diameter of pitch circle and number of bolts, however, to be used on all pressures between 100 and 300 pounds, which I have termed "A Proposed Universal Standard Flange List," the intention being to use the list as designed for pressures between 200 and 300 pounds, and then to modify the size of the bolts used for pressures between 100 and 200 pounds on all pipes above 4 inches diameter, so that we would never use less than a $\frac{5}{8}$-inch bolt within the range of sizes given.

It has been my aim to produce a list whereby various methods may be employed for attaching flanges to pipes with-

out affecting in any manner the principal points of advantage sought for, and in order to fully understand my meaning in this direction I have had prepared sections of 10-inch pipe with their flanges attached as shown.

It will be noticed that I have made a comparison of the single-riveted flange with the pipe flared at the outer end to an angle of 20 degrees. (See sheet No. 4.) I have also shown the double riveting, such as we have used on all our merchant ships, with the pipe caulked inside and outside, similar to that of a boiler-shell joint. (See sheet No. 5.)

Furthermore, I have prepared the double-riveted joint as used by the U. S. Navy, wherein the end of the pipe is pened over into a recess in the face of the flange. (See sheet No. 6.) In all these designs I have kept the section where the bolt passes through the same throughout. Knowing this section to be more in accordance with mechanical principles than the parallel section commonly employed, caused me to adopt this form, and I have had numerous samples made and tested, so as to prove my statements.

Furthermore, I might mention that in order to be certain as to what could be accomplished in the direction of expanded joints, I made several samples of steel pipe 20 inches in diameter and one-half inch thick, and the work done was such that the Committee of Science and Arts of the Franklin Institute awarded me the Elliott Cresson Gold Medal for my work in this new field of engineering. In all cases where expanded joints are made, I am certain that, if necessary, we can burst the pipe without in any way affecting the expanded joint when properly made.

Furthermore, the expanded joint is at least 25 per cent. stronger than the average form of riveted joint, such as shown in the accompanying prints.

It is also absolutely impossible to have a leak between the flange and the pipe when so expanded, for I have made numerous tests wherein I have heated both the pipe and the flange red hot, and then plunged the same in water, and afterwards placed them under a test pressure of 5,000 pounds per square

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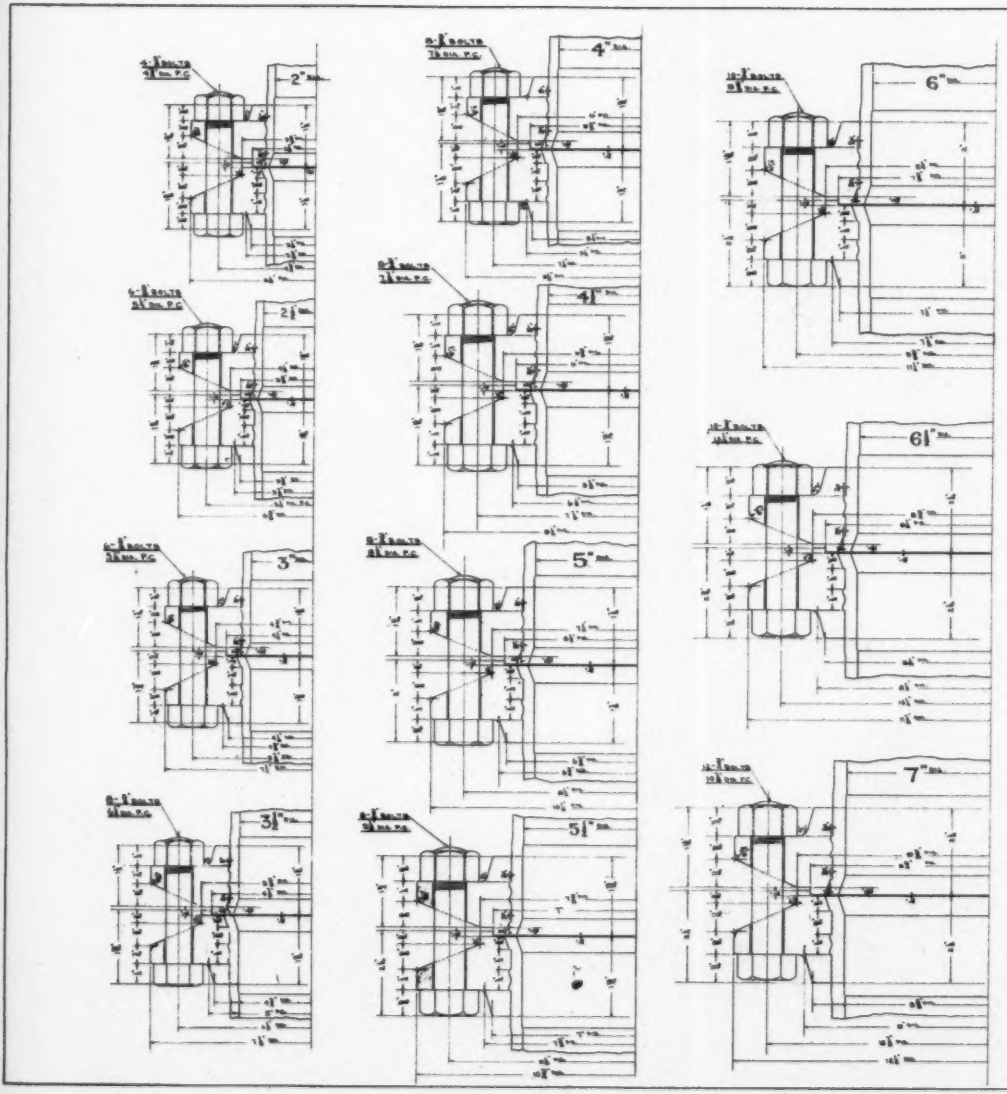
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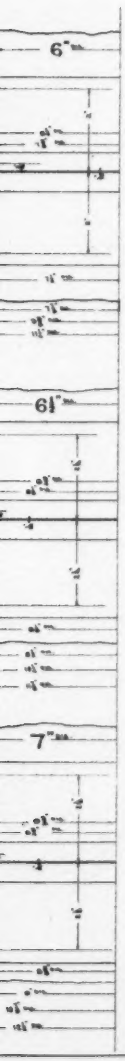
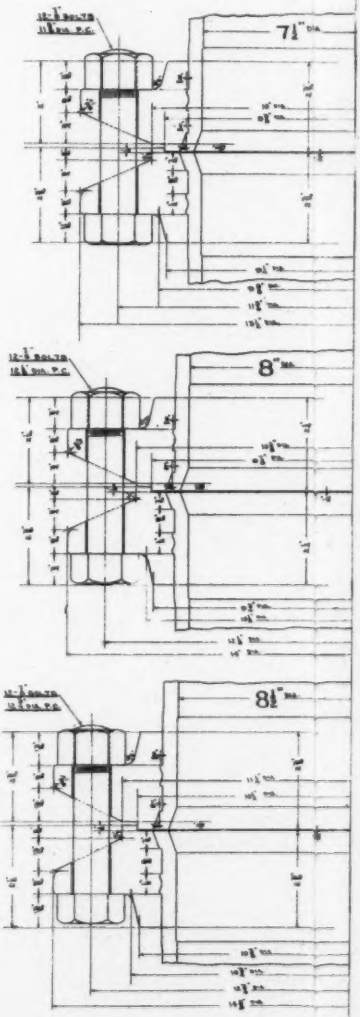
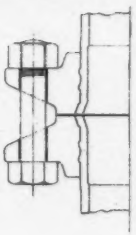



NOTE.—

THIS LIST IS SO ARRANGED THAT FLANGES HAVING A DEPTH OF BOSS AT THE BACK SUITABLE FOR EITHER SINGLE OR DOUBLE RIVETTING CAN BE EITHER CONNECTED TO, OR THEY CAN BE USED IN LIEU OF THE EXPANDED PIPE TYPE WHEN SO REQUIRED, WITHOUT AFFECTING THE PITCH CIRCLE, DIAMETER OF FLANGE, OR THE SHAPE OF THE SECTION.

THEY ARE ALSO SUITABLE FOR CONNECTING ANY ABRADING CASTING OR VALVE WITHOUT CHANGE OF DIMENSIONS.



NOTE.—

FOR PRESSURES BELOW 175 LBS. THE MALE AND FEMALE JOINT MAY BE OMITTED AS SHOWN ABOVE.

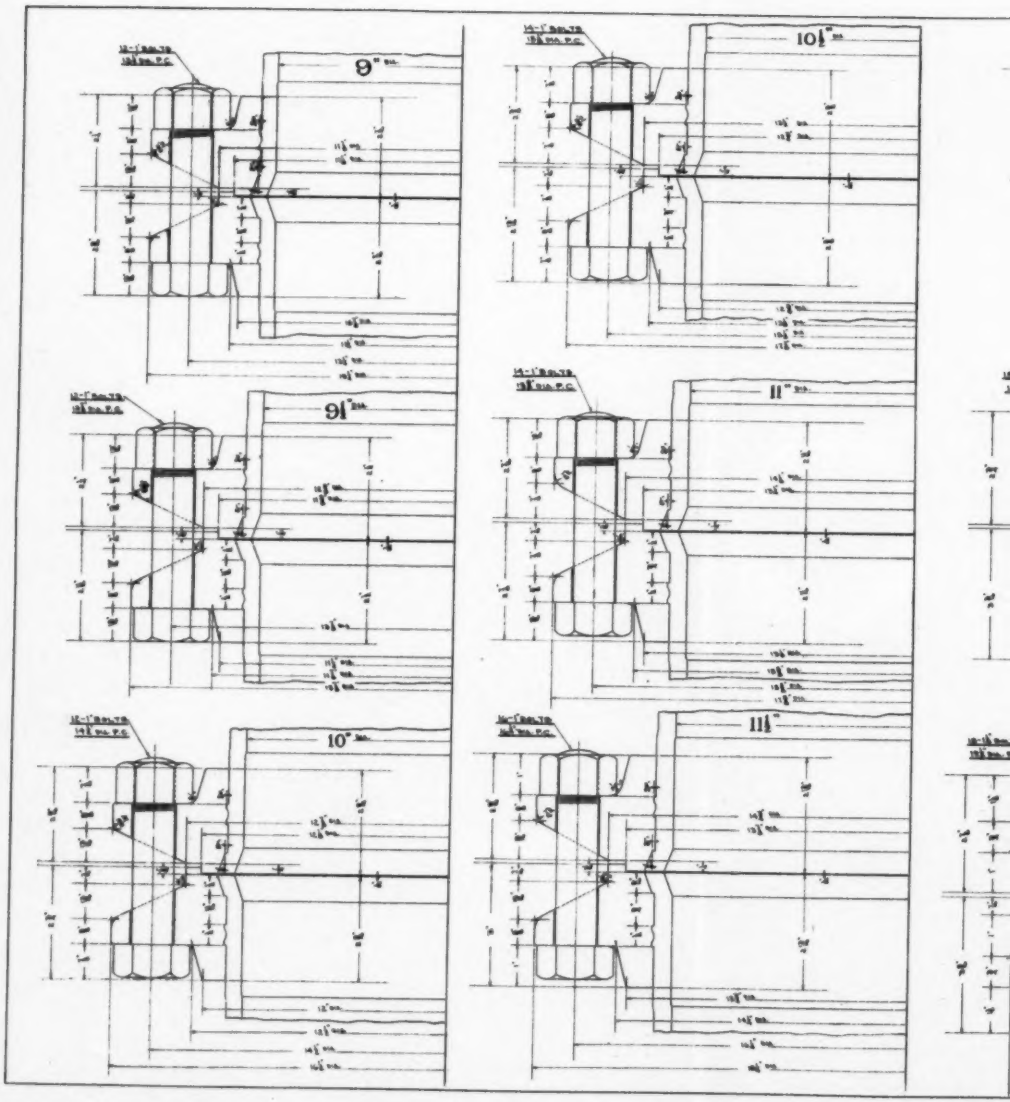
ALL PIPES WITH THEIR FLANGES CONNECTED WILL BE TESTED TO 2½ TIMES THE WORKING PRESSURE.

THESE FLANGES ARE TO BE USED ON COPPER, IRON AND STEEL PIPES. ALL FLANGES TO BE OF GOOD FORGED IRON OR STEEL OR STEEL CASTINGS.

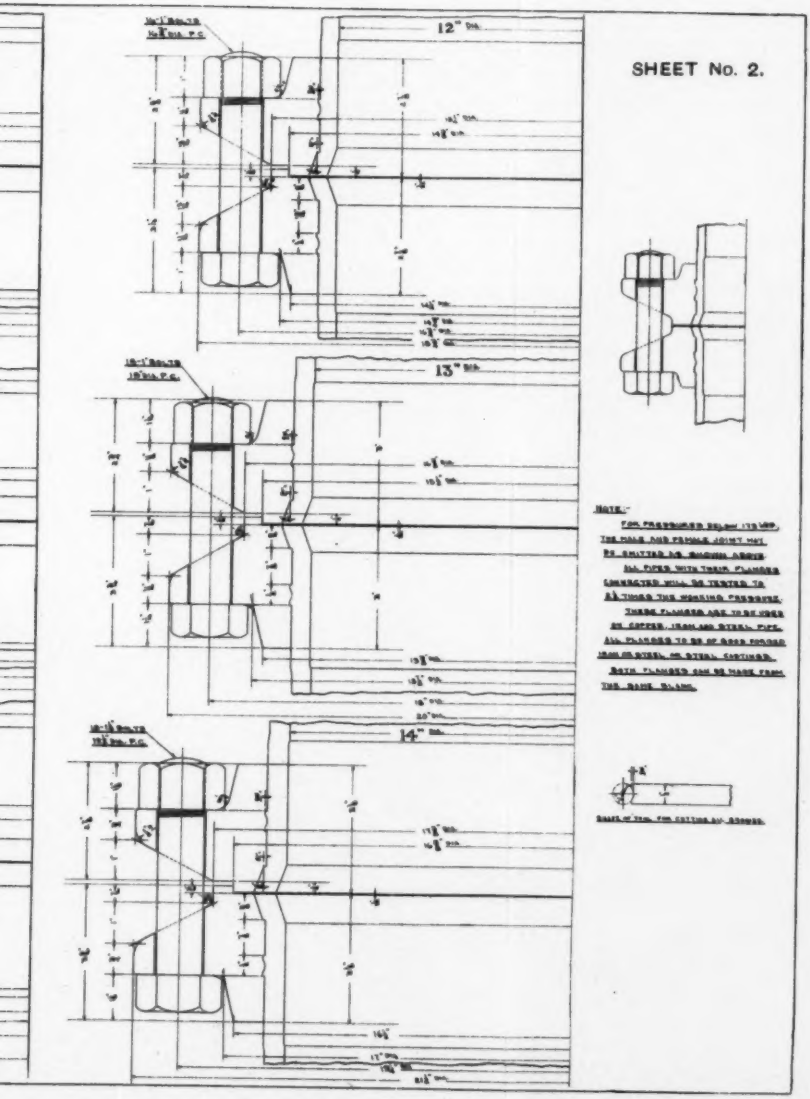
BOTH FLANGES CAN BE MADE FROM THE SAME BLANK.

SHAPE OF IRON FOR CUTTING ALL SPACES

SHEET No. 1.



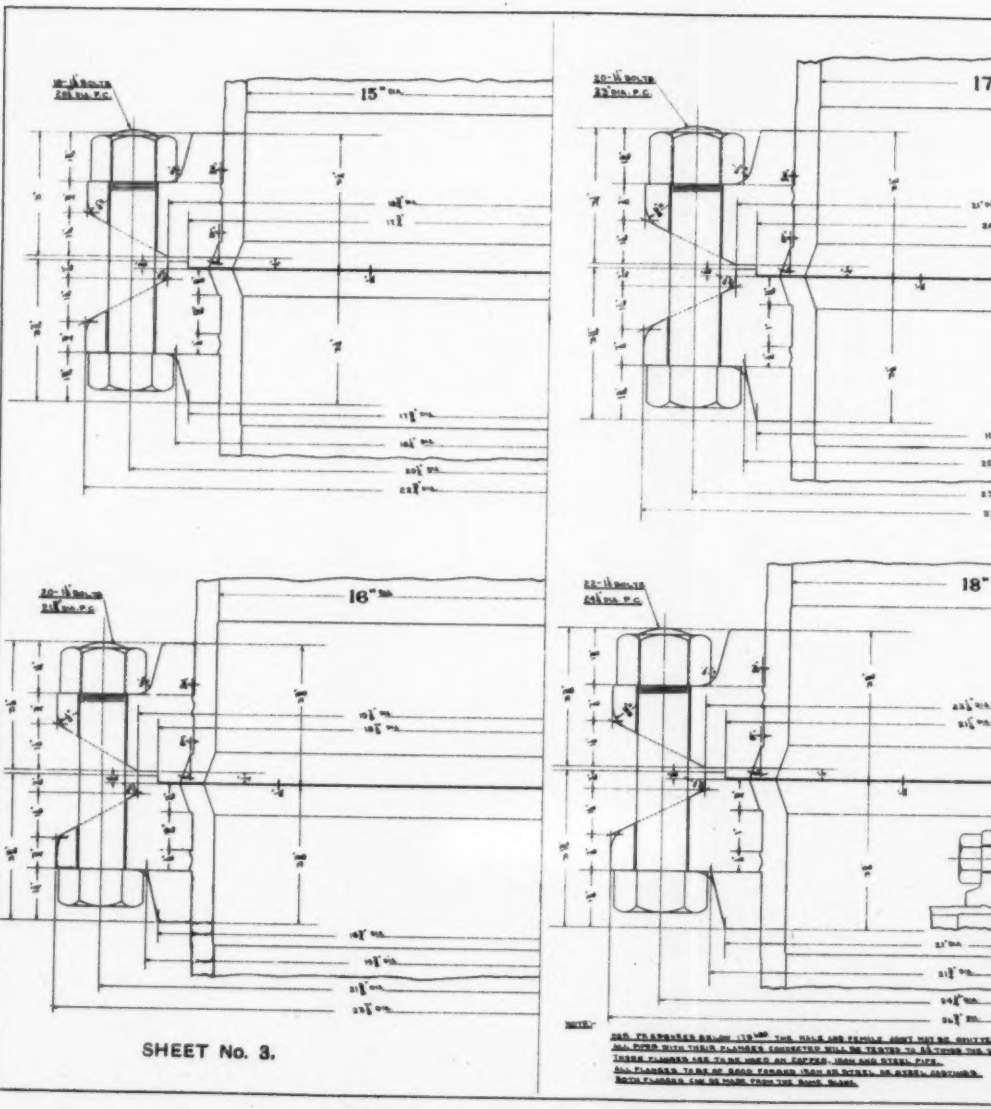
SHEET No. 2.



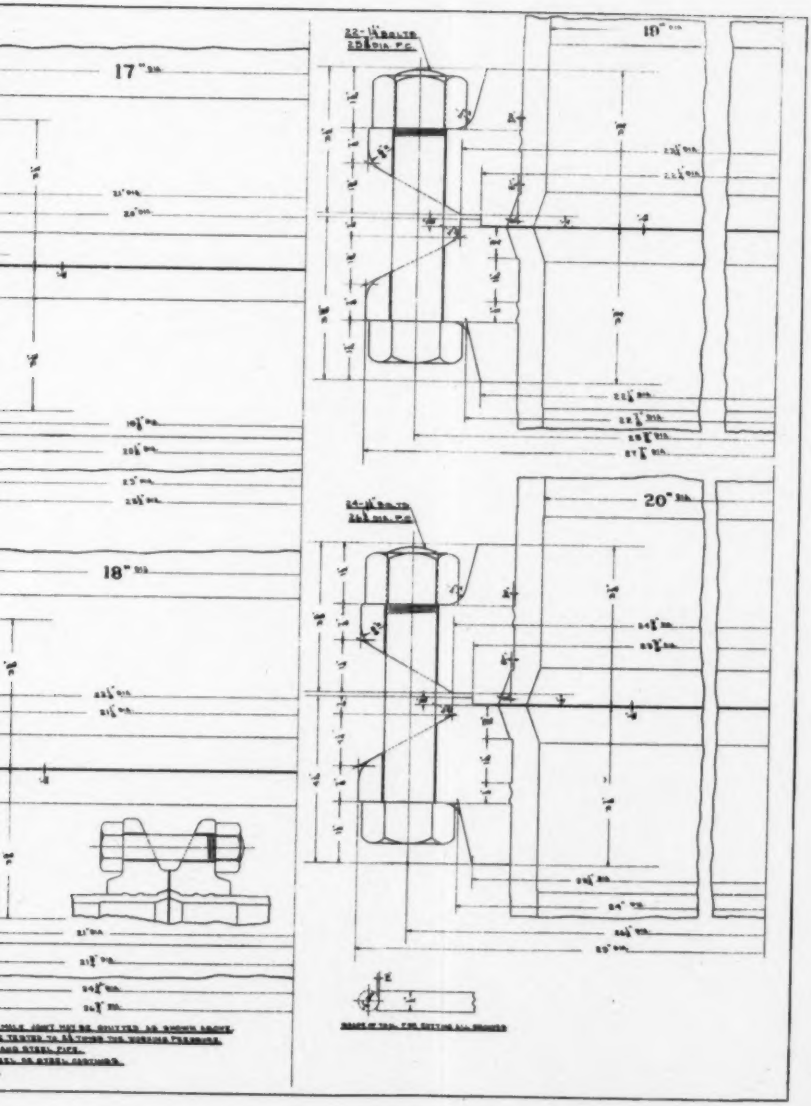
NOTE:-

FOR PRESSURES EXCEEDING 175 LBS. THE MALE AND FEMALE JOINT MAY BE SHUTTED AS SHOWN ABOVE ALL PIPES WITH THEIR FLANGES CONNECTED WILL BE TESTED TO 1½ TIMES THE WORKING PRESSURE. THESE FLANGES ARE TO BE USED ON COPPER, IRON AND STEEL PIPE. ALL FLANGES TO BE OF GOOD POWDER IRON OR STEEL OR STEEL CASTINGS. BOTH FLANGES CAN BE MADE FROM THE SAME BLANK.

SHAPE OF TOOL FOR CUTTING OUT GROOVE.



SHEET No. 3.


17"
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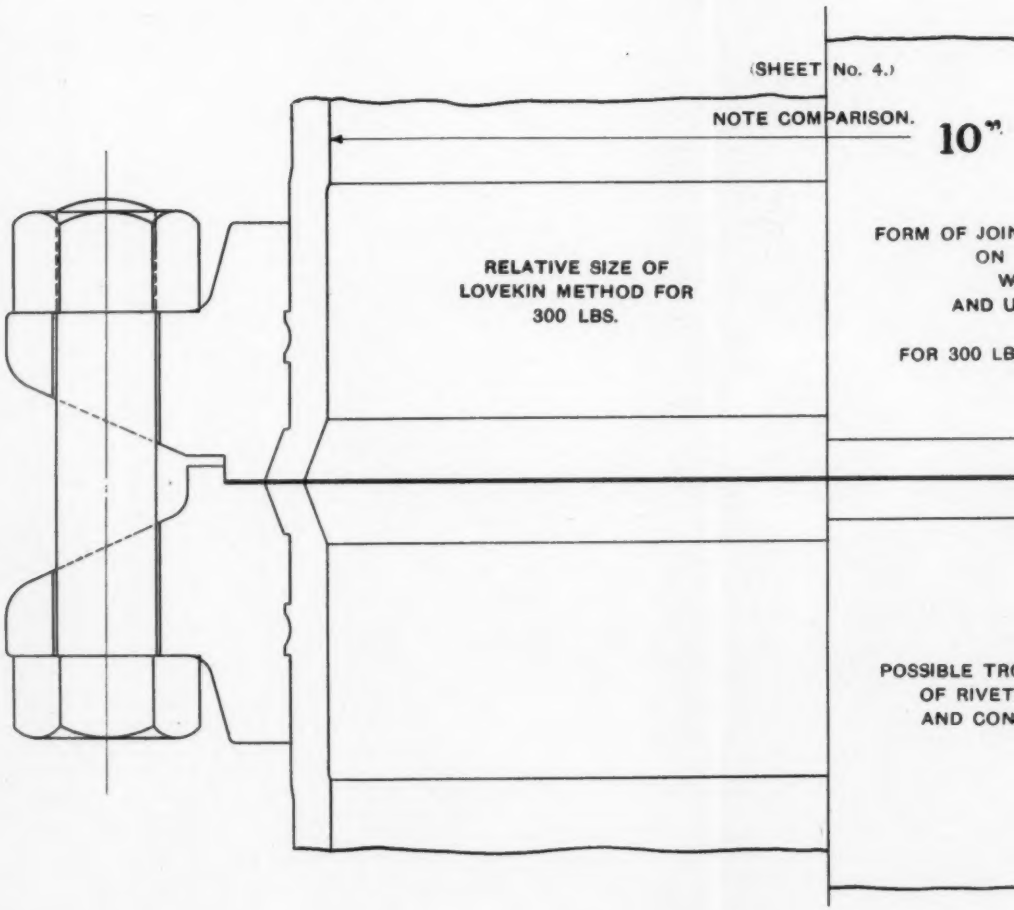
(SHEET No. 4.)


NOTE COMPARISON.
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RELATIVE SIZE OF
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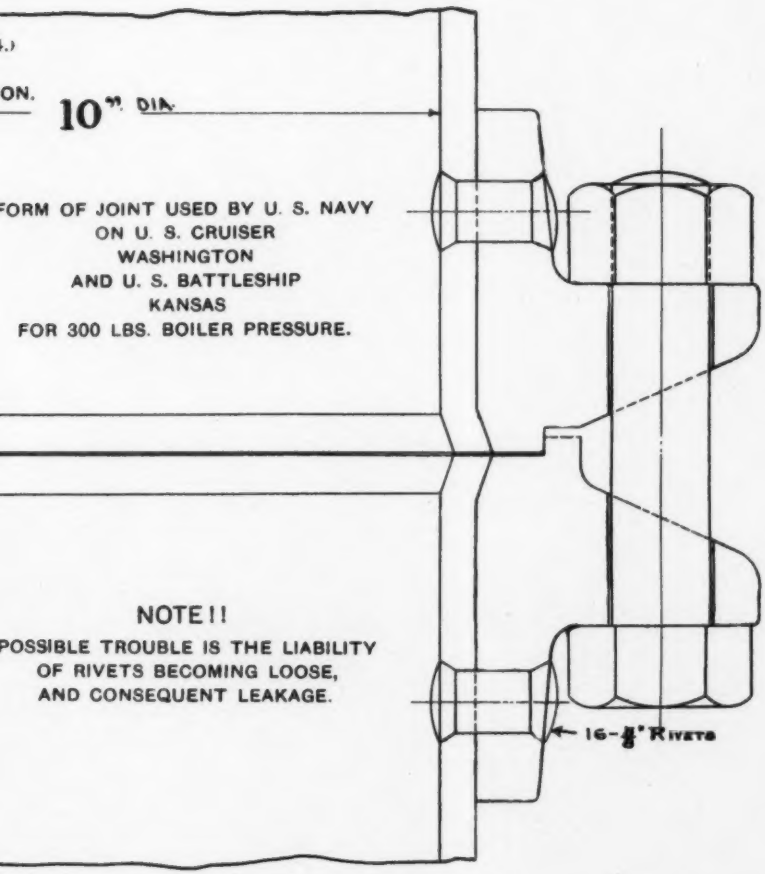
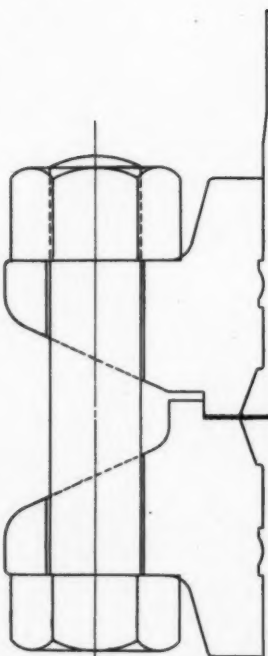


FORM OF JOINT USED BY U. S. NAVY
ON U. S. CRUISER
WASHINGTON
AND U. S. BATTLESHIP
KANSAS
FOR 300 LBS. BOILER PRESSURE.

NOTE!!

POSSIBLE TROUBLE IS THE LIABILITY
OF RIVETS BECOMING LOOSE,
AND CONSEQUENT LEAKAGE.



(SHEET No. 5.)

NOTE COMPARISON.

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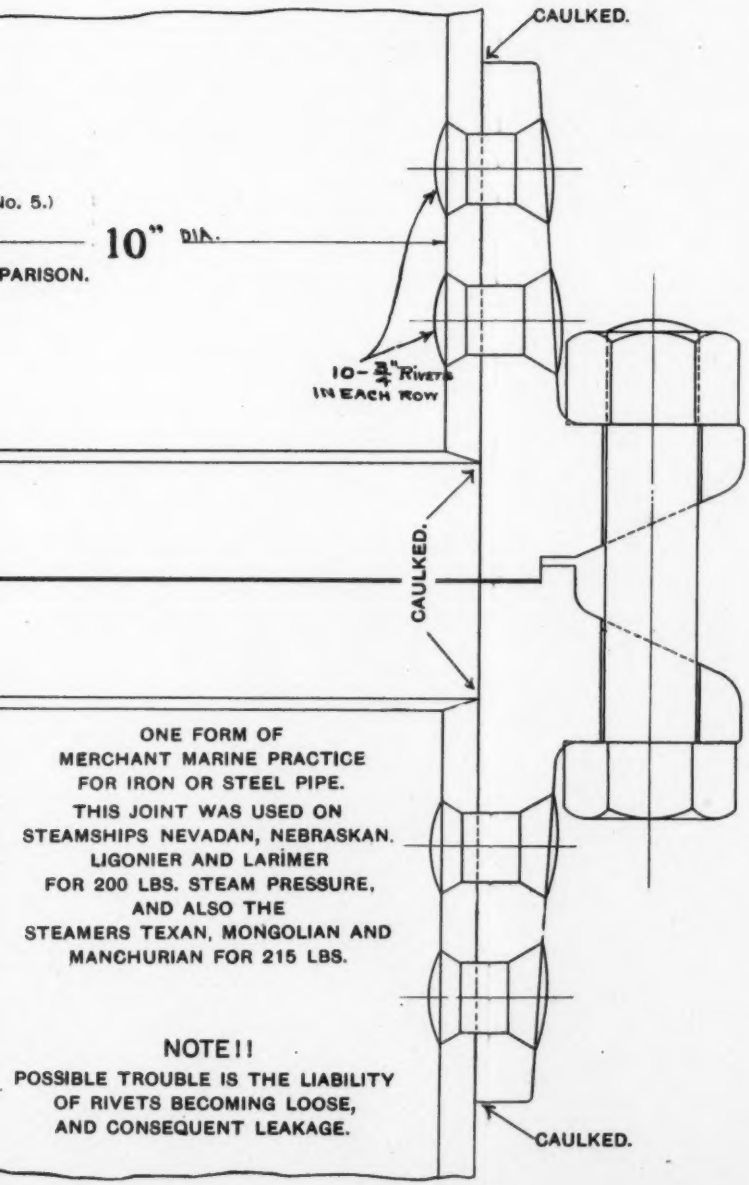


(No. 5.)

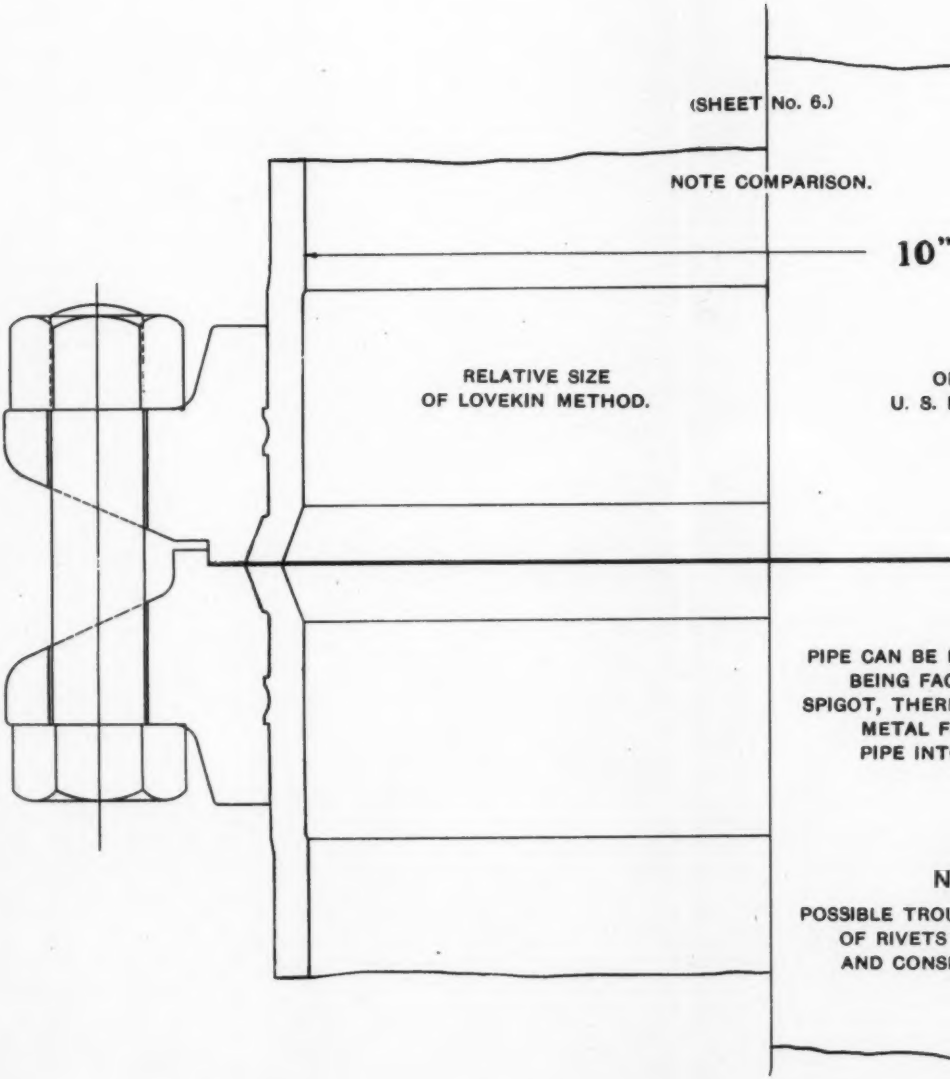
PARISON.

ONE FORM OF
MERCHANT MARINE PRACTICE
FOR IRON OR STEEL PIPE.

THIS JOINT WAS USED ON
STEAMSHIPS NEVADAN, NEBRASKAN,
LIGONIER AND LARIMER
FOR 200 LBS. STEAM PRESSURE,
AND ALSO THE
STEAMERS TEXAN, MONGOLIAN AND
MANCHURIAN FOR 215 LBS.

NOTE!!

POSSIBLE TROUBLE IS THE LIABILITY
OF RIVETS BECOMING LOOSE,
AND CONSEQUENT LEAKAGE.


(SHEET No. 6.)
NOTE COMPARISON.
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RELATIVE SIZE
OF LOVEKIN METHOD.
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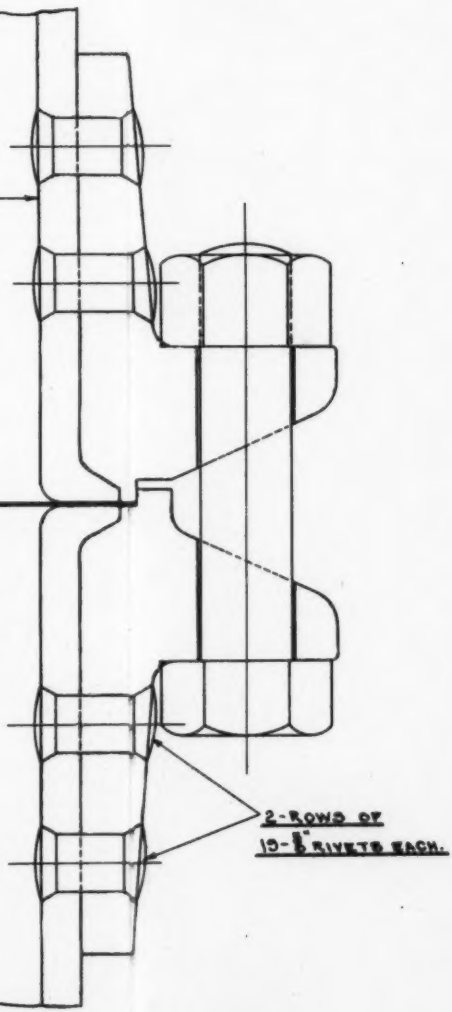
ONE FORM OF
U. S. NAVY PRACTICE.



IPE CAN BE PENED OVER BEFORE
BEING FACED OFF AT THE
PIGOT, THEREBY ALLOWING MORE
METAL FOR HAMMERING
PIPE INTO THE FLANGE.

NOTE!!

OSSIBLE TROUBLE IS THE LIABILITY
OF RIVETS BECOMING LOOSE,
AND CONSEQUENT LEAKAGE.

inch, without any sign of leakage whatever. *This will be apparent to all when it is considered that the ends of the pipes themselves form the joint and any additional clamping effect produced by the bolts has a tendency to pull the pipe closer to the flange, in direct proportion to the angle of flare at the end of pipe, the small shoulder at the end preventing the pipe being forced out of the flange.* As stated previously, my aim has been to produce a flange list whereby any of the methods shown may be employed without affecting the principal points of advantage sought for.

Furthermore, we all realize what a great advantage would be had in a universal flange list, and in order to accomplish this purpose it will require the united efforts of both the shipbuilders, the Navy Department and the Bureau of Commerce and Labor (Steamboat Inspection Service). This latter Bureau I have satisfied, and any of the forms shown herewith meets with their approval, according to the latest rules and regulations (1905 edition).

In order that there should be no one particular form of flange connection considered in this matter I have, as before stated, considered all of the joints used in marine work at the present time, with the exception of the welded flange. My reason for not considering this is on account of the enormous expense necessary to make the same, and also the inability of using a flange welded to the pipe to its greatest advantage, such advantage being a very small diameter of flange and also the ability to use a small pitch circle, due to not having any boss on the back of the flange; *this type of joint could never be employed as a universal standard*, inasmuch as any standard flange list adopted must have the following characteristics, viz:

We should be able to join a pipe to a valve or similar casting, when made of cast iron, the thickness of metal in such valves or castings being always in excess of the thickness of material in the pipe itself; we should also provide for pipes having their flanges rivetted to the pipe. Consequently we must have a pitch circle that will readily conform to the above,

so that the nuts or heads of the bolts may clear the fillet in the casting. All flanges submitted by me have had these points fully considered, and the advantages sought are as follows:

First. A uniform outside diameter of pipe flange, for pressures between 100 and 300 pounds.

Second. A uniform pitch circle for bolting flanges together.

Third. A sufficient number of bolts to be equal in strength to the bursting pressure of the pipe.

Fourth. A flange section fully as strong as the bolt section in all cases, so designed that the weight of same is reduced to a minimum.

Fifth. To have all bolts entirely outside of the gasket, thereby avoiding troublesome leakage through bolt holes, such as is often encountered.

Sixth. That the areas of the gaskets between flanges may be subjected to a uniform pressure per square inch in all cases by means of the number of bolts used, said total pressure on gaskets being in excess of the tendency to pull joints apart (due to the internal pressure in the pipe) without subjecting the bolts to a stress of over 5,000 pounds per square inch under working pressure.

This feature of the gasket area is a very important one, and the design submitted requires the mimimum amount of material that could be desired for such purpose, thus securing a great saving to both the builder and owners in the supply and renewal of these gaskets.

Seventh. The flanges, as shown, may be used for either copper or steel pipe, and can be made of steel or iron for copper pipe when not subjected to corrosion (such as pipes in the bilges of ships or in any other locality where galvanic action plays an important part), therefore *eliminating the practice of brazing the copper pipe to the flange*, which, from results shown, is positively not a necessity, but rather a source of danger. Where necessary or advisable to use brazing metal or bronze flanges, they may be of the same sectional area of the steel flanges shown, inasmuch as these steel flanges have been designed for steel pipe, and the same sections may be made of brazing metal

for copper pipe, and bear practically the same relation as the steel flanges for the steel pipe.

The third and fourth paragraphs are the ones I have given great consideration, and are the basis of all my work. I am not a believer in that practice of engineering which permits the use of varying factors of safety for the same structure when subjected to all kinds of shocks, such as water hammers, etc. We should certainly have the pipe with a certain factor of satety and then make the flange connection with approximately the same factor. In all of my work, as will be clearly shown in Fig. No. 7, I have taken the bursting strength of the pipe as the basis for all calculations relating to these joints. A full set of calculations appear on pages following, which clearly indicates the manner in which each and every one of these flanges have been designed.

It will be noted that for pressures above 175 pounds I have used the male and female type of joint; for pressures below 175 pounds a plain joint is used, such as shown in the drawings of proposed universal flange list. While, as stated before, any of our well-known methods may be employed in connection with this universal standard flange list, I would state that the advantages of the expanded pipes as shown are as follows:

No heating, no brazing, no riveting, *no leakage such as is often encountered through rivet holes or between pipe and the inside of flange*, no crystallization. Joint may be designed to exceed the strength of the pipe. By the omission of rivet holes the pipe is about 25 per cent. stronger than riveted pipe.

Furthermore, I have made many samples wherein from actual tests the pipe showed an increase in strength due to the cold rolling it received of about 15 per cent.

In conclusion, I would state that with a flange constructed on this principle the tightness of joints at the gasket will not be affected by the pitch of the bolts, inasmuch as the great depth of flange over the gasket is not susceptible of any deflection whatever, and it is therefore impossible to blow a gasket out, due to the deficiencies such as occur in the older forms

of flange connections. All that is needed in a design of the character shown is a sufficient number of bolts to enable the flanges to be drawn together to such an extent as to produce a pressure on the gasket in excess of the tendency to pull apart, and a section of flange that will not be unduly strained under such pressure. The depth of the flange through the bore must be sufficient for either single or double riveting or expanding, and the depth of bores in the expanded joints shown herewith are the results of numerous tests made by me, and are what I consider the least possible for safety.

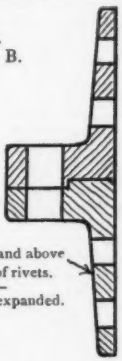
I enclose herewith a table (marked Fig. No. 8) showing all the various stresses throughout, of this proposed universal flange list, which gives the various proportions throughout in a condensed form, and I trust that this matter will be considered of sufficient importance to give the same thorough consideration, knowing, as I do, that this is a matter which has been under consideration for some time, and believing that a flange list for the Navy Department should not differ in any way from a flange list for the merchant marine, causes me to introduce this subject in the manner I have done.

A comparison of these flanges with those in use by the U. S. Navy Department will show very clearly the great saving in weight over the above-mentioned flanges, and in addition will also convince one of their superiority over the former standard flanges as regards strength. (See list of comparative weights, Tables No. 9 and 10.)

[NOTE.—A full and complete discussion is invited on this subject, and it is hoped that the various shipbuilders and engineers will give this matter their consideration on account of its great importance, addressing their remarks to the Secretary-Treasurer.]

Table IX.

COMPARISON OF WEIGHTS BETWEEN "LOVEKIN UNIVERSAL" PROPOSED LIST (COLUMN A) AND THE U. S. NAVY DEPARTMENT, BUREAU OF STEAM ENGINEERING, STANDARD, WEIGHT OF ONE PAIR, MALE AND FEMALE, ALL CALCULATED OR SCALED; NO DRILLING, BUT BORED AND FACED TO WITHIN ONE-SIXTEENTH INCH OF DRAWING DIMENSIONS.

Diameter.	Lovekin. Column A.	Navy. Column B. 
Inches.	*Pounds.*	*Pounds.*
2	12.5*	12.5†
2½	14.6*	17.†
3	17.3*	25.†
3½	20.8*	28.†
4	23.9*	34.†
5	35.6*	44.†
6	45.1*	72.8*
7	58.8*	83.7*
8	73.1*	103.3*
10	105.7*	135.4*
12	144.7*	195.*
15	221. *	275.*

As they come from the die.

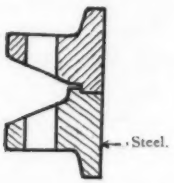
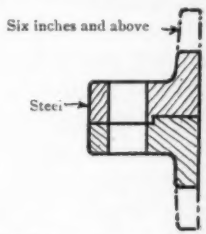
*Estimated. †Scaled.

NOTE.—The Lovekin flange is very much stronger than the Navy type as shown, owing to the thickness of flange at the bolt section. Flanges are less difficult to make, and thus a great saving is obtained in price, owing to the omission of the great depth of boss.

Table X.

COMPARISON OF WEIGHTS BETWEEN "LOVEKIN UNIVERSAL" PROPOSED LIST (COLUMN A) AND THE FLANGES USED ON THE *WASHINGTON* AND *KANSAS* (COLUMN B), FOR 300 POUNDS GAUGE PRESSURE.

Weight of one pair (male and female) faced and straight bored to within one-sixteenth inch of drawing dimensions, calculated or scaled without any bolt holes or rivet holes being considered.

Diameter.	Column A. 	Column B. On six inches and over collar is lengthened and a single row of rivets used. 
Inches.	*Pounds*	*Pounds.*
2	12.5*	12.5†
2½	14.6*	17.†
3	17.3*	25.†
3½	20.8*	28.†
4	23.9*	34.†
5	35.6*	44.†
6	45.1*	57.†
7	58.8*	68.*
8	73.1*	86.†
10	105.7*	115.8*
12	144.7*	166.†
15	221. *	237.*

As they come from the die.

* Estimated. † Scaled.

NOTE.—The Lovekin flange is very much stronger than the ones used on the *Washington* and *Kansas*, owing to the thickness of flange at the bolt section.

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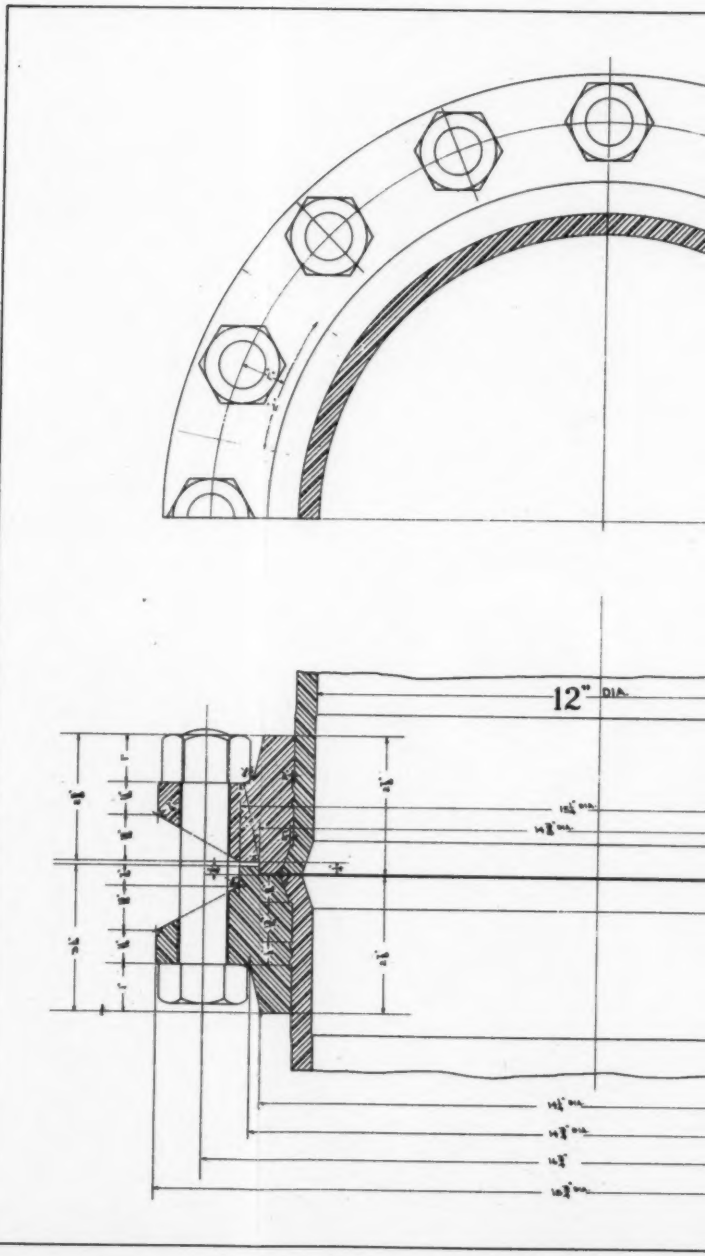
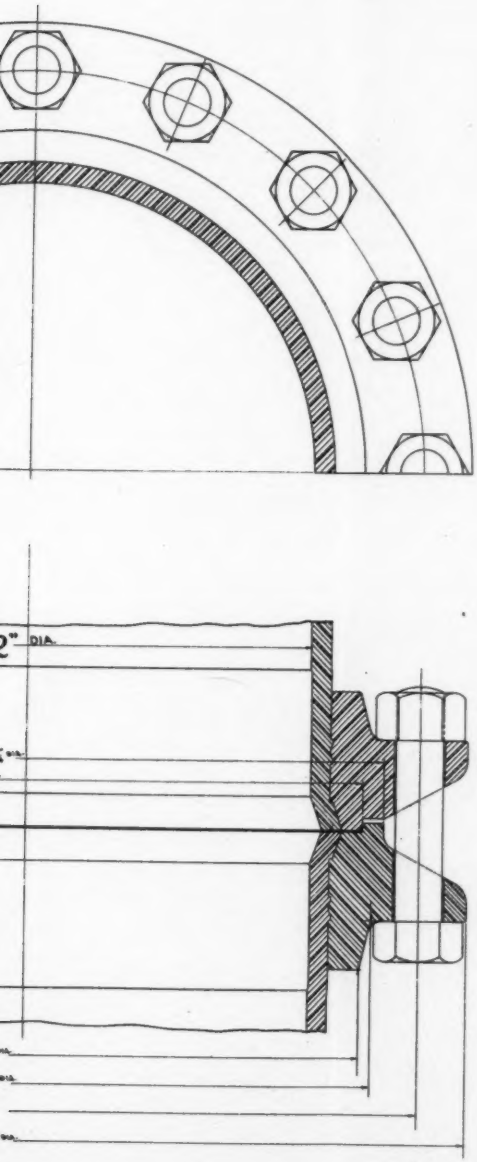

12" DIA.

FIG. No. 7.



CALCULATIONS FOR A 12-INCH FLANGE.

See Figure No. 7.

Inside diameter of pipe, nominal, 12 inches.

$$T = \frac{P \times D}{10{,}000} + \tfrac{1}{8} \text{ inch, U. S. Navy rule.}$$

$P = 265$ pounds per square inch.

$D = 12$ inches.

$$T = \frac{265 \times 12}{10{,}000} + \tfrac{1}{8} \text{ inch} = .443, \text{ say } \frac{29}{64} = .4503.$$

Bursting pressure $P = \dfrac{2TS}{D}$.

Assume $S = 50{,}000$ ultimate;

Hence, $P = \dfrac{2 \times \frac{29}{64} \times 50{,}000}{12} = 3{,}780$ pounds per square inch;

Factor of safety for 300 pounds $= \dfrac{3{,}780}{300} = 12.6$.

Use 16 bolts 1 inch diameter. Area at root of thread $= .55$ square inches.

Assume that the end pull, corresponding to the bursting pressure, will be over the area of the pipe to the inside diameter of the gasket.

Area for $12\frac{1}{2}$ inches diameter $= 122.72$ square inches.

Pull on bolts $= 122.72 \times 3{,}780 = 463{,}800$ pounds.

Pull on each bolt $= \dfrac{463{,}800}{16} = 29{,}100$ pounds.

Stress per square inch at root of thread $= \dfrac{29{,}100}{.55} = 53{,}000$ pounds.

Outside diameter of gasket $14\frac{5}{8}$ inches $= 162.3$ square inches, area;

Inside diameter of gasket $12\frac{1}{2}$ inches $= 122.72$ square inches, area.

Area of gasket $= 39.58$ square inches.

When the joint is tightened up, assume that it is compressed to a pressure of 1,000 pounds per square inch, as this may be necessary to imprint the flange faces thoroughly. The resultant stress on the bolts would be $\dfrac{39.58 \times 1{,}000}{16 \times .55} =$ 4,500 pounds per square inch.

Under working conditions, at 300 pounds pressure in the pipe, assume that this same pressure is maintained over the gasket area, so there can be no chance for the steam to enter between the faces in contact, and thus invite a leak.

Hence, working stress on bolts $= \dfrac{162.3 \times 300}{16 \times .55} = 5{,}500$ pounds per square inch.

CALCULATIONS FOR STRENGTH OF FLANGE ON THE SECTION AB TO RESIST THE MOMENT OF THE BOLT WHEN THE PIPE IS SUBJECT TO A BURSTING PRESSURE.

Assume that each bolt acts on its own proportionate part of the flange; that is, the flange is divided into sixteen beams, fixed at one end and carrying the pressure of the bolt at the other.

F = pressure of one bolt due to the bursting pressure in the pipe = 29,100.
L = distance of application from neutral axis = $1\frac{1}{32}$ inches.
Hence, moment = $F \times L$.
Moment of resistance for this beam = $\frac{sbh^2}{6}$.
h = depth of beam (length of AB) = $1\frac{1}{16}$ inches.
s = maximum fiber stress in the section.
b = breadth of beam measured along the circumference of neutral axis = 2.88 inches.

$$F \times L = \frac{sbh^2}{6}.$$

Hence, $S = \frac{6FL}{bh^2} = \frac{6 \times 29{,}100 \times 1\frac{1}{32} \text{ inches}}{2.88 \times (1\frac{1}{16})^2} = 22{,}000$ pounds per square inch.

Thus the flange cannot be bent by the bolts when the pipe is subject to its bursting pressure, on account of our keeping the stress below the elastic limit of the material in the flange.

Stresses on Grooves When the Pipe is Subject to Bursting Pressure.

Area of shoulder, $\frac{3}{64}$ inch deep.
$13\frac{1}{16}$ inches outside diameter = 133.9 square inches area.
$12\frac{3}{32}$ inches inside diameter = 131.9 square inches area.
= 2. square inches net area.

The projected area of the curved shoulder is also equivalent to 2 square inches area; hence the bearing area of the grooves = $2 \times 2 = 4$ square inches.

The projected area of the flare is the most effective bearing surface against end pull on the pipe.
$13\frac{1}{4}$ inches outside diameter = 143.1 square inches area.
$12\frac{3}{32}$ inches inside diameter = 131.9 square inches area.
= 11.2 square inches net area.
4
Hence the total bearing area = 15.2 square inches.

Crushing Stress Due to Bursting Pressure in the Pipe.

$$\frac{122.7 \times 3{,}780}{15.2} = 30{,}500 \text{ pounds per square inch.}^*$$

Depth of shearing area = depth of grooves + depth of flare = $\frac{1}{4}$ inch + $\frac{3}{16}$ inch = $1\frac{1}{16}$ inch.
Circumference of shearing area for $12\frac{3}{32}$ inches diameter = 40.8 inches.
Shearing area = $40.8 \times 1\frac{1}{16}$ inches = 43.4 square inches.

Shearing stress due to *bursting pressure* = $\frac{122.7 \times 3{,}780}{43.4} = 10{,}700$ pounds per square inch.†

* This stress could be at least 50,000 pounds.
† This stress could be at least 40,000 pounds.

IMPORTANT NOTE.

The principal bearing surface in this form of joint is the projected area of the flared portion at the end of the pipe. The grooves in the bore of the flange, however, have a very important function with this type of joint, serving to retain the flange in a positive position with relation to the end of the pipe, and also partially relieving the shearing and crushing stresses at the ends.

The complete filling of these grooves is not essential to the effectiveness of this type of joint.

See cut No. 7A, showing diagram of the resolution of forces, attached herewith.

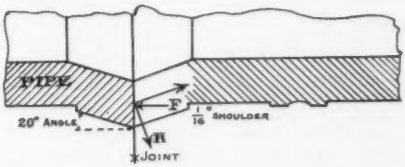


Fig. 7a.

Diagram of Resolution of Forces in Lovekin Method of Making Flange Joints.

Note the outward reaction of flared pipe due to end pressure.

R = reaction.
F = end pressure.
$R = F \sin 20°$.
$R = .342\ F$.

The coupling also tends to close in on the flare of the pipe, due to the moment of the bolts. This maintains the close contact between the pipe and the flange.

The dimensions of pipe flanges for universal standard follow in table, Fig. No. 8.

The method of calculating the pull imposed on the bolts by the bursting pressure in the pipe may be open to question. It can be argued that it is better to consider the pressure over the whole area of the pipe and gasket, instead of taking it to the inside diameter of gasket only.

In accordance with this argument that the pressure is distributed over the whole area, it will be well to give a column of figures showing the working stress on the bolts due to 300 pounds pressure over the entire surface; the table herewith appended gives such.

Stress in Bolts due to 300 Pounds Pressure Over the Entire Area of Pipe and Gasket.

Diameter of pipe.	Stress in bolts.	Diameter of pipe.	Stress in bolts.	Diameter of pipe.	Stress in bolts.
Inches.	*Pounds.*	*Inches.*	*Pounds.*	*Inches.*	*Pounds.*
2	2,850	7	4,980	12	5,540
2½	2,550	7½	4,100	13	5,720
3	3,510	8	4,560	14	5,220
3½	3,460	8½	5,150	15	5,940
4	4,210	9	4,420	16	6,050
4½	3,510	9½	4,820	17	5,280
5	4,120	10	5,250	18	5,350
5½	4,780	10½	4,970	19	5,930
6	4,540	11	5,360	20	6,000
6½	4,870	11½	5,150		

All bolts to be of either good mild steel or the best grade of wrought iron. No special material required in any case.

Steel of 60,000 tensile strength, iron of 45,000 tensile strength have ample safety in all cases.

XU

DIMENSIONS OF PIPE FLANGES FOR UNIVERSAL ST

100 LBS TO 300 LBS PRESSURE

FOR PRESSURES BELOW 175 LBS THE MALE AND FEMALE JOINT MAY BE OMITTED.

NOMINAL INSIDE DIAMETER OF PIPE.	T. THICKNESS OF PIPE FOR $T = \frac{P \times D}{10{,}000} + \frac{1}{16}$		B. OUTSIDE DIA. OF FLANGE	INSIDE DIA. OF FLANGE	K. OUTSIDE DIAM. OF GASKET		P. DIAMETER OF SPIGOT		S.	O. DEPTH OF FLANGE FROM JOINT FACE		H. DEPTH OF HUB.		F. MINIMUM DIAM. OF HUB.	E. MAXIMUM DIA. OF HUB		L.	C. DIAM. OF CENTRAL CIRCLE	NUMBER OF BOLTS.	DIAMETER OF STEEL BOLTS		R.	
	BY RULE	APPROX.				DEPTH OF FACE IN GASKET		RATIO TO T			RATIO TO T	T	A			THICKNESS OF METAL IN FLANGE							
2	.178	$\frac{7}{16}$	$6\frac{1}{2}$	$2\frac{7}{16}$	$3\frac{1}{2}$	1.2	$3\frac{5}{8}$	$\frac{1}{4}$	1.3	$\frac{3}{8}$	$1\frac{1}{2}$	8	$\frac{3}{4}$	.2	3.	$3\frac{1}{2}$	1.8	$3\frac{3}{8}$	$\frac{1}{8}$	$\frac{3}{8}$	$1\frac{1}{2}$	4	$\frac{3}{4}$	$\frac{1}{4}$	1.3
$2\frac{1}{2}$	.191	$\frac{3}{16}$	$6\frac{1}{2}$	$2\frac{15}{16}$	$3\frac{1}{2}$	1.2	$4\frac{1}{8}$	$\frac{1}{4}$	1.3	$\frac{3}{8}$	$1\frac{1}{2}$	8.3	$\frac{7}{16}$	.23	.33	$3\frac{1}{4}$	1.8	$3\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{8}$	$1\frac{1}{2}$	$5\frac{1}{2}$	6	$\frac{3}{8}$	$\frac{1}{4}$	1.3
3	.208	$\frac{13}{64}$	$7\frac{1}{2}$	$3\frac{13}{16}$	$4\frac{1}{2}$	1.3	$4\frac{7}{8}$	$\frac{1}{4}$	1.3	$\frac{3}{8}$	$1\frac{5}{8}$	8.	$\frac{7}{16}$	.22	.33	$4\frac{1}{8}$	1.6	$4\frac{3}{8}$	$\frac{1}{8}$	$\frac{3}{8}$	1	$5\frac{7}{8}$	6	$\frac{5}{8}$	$\frac{1}{4}$	1.2
$3\frac{1}{2}$	.218	$\frac{7}{32}$	$7\frac{7}{8}$	4	$4\frac{7}{8}$	1.3	$5\frac{3}{8}$	$\frac{1}{4}$	1.4	$\frac{3}{8}$	$1\frac{11}{16}$	7.7	$\frac{1}{2}$	.23	.4	$4\frac{7}{8}$	1.7	5	$\frac{1}{8}$	$\frac{7}{16}$	1	$6\frac{1}{2}$	8	$\frac{5}{8}$	$\frac{5}{16}$	1.4
4	.231	$\frac{15}{64}$	$8\frac{1}{2}$	$4\frac{13}{16}$	$5\frac{3}{8}$	1.2	6	$\frac{5}{16}$	1.3	$\frac{3}{8}$	$1\frac{3}{4}$	7.5	$\frac{5}{8}$	.21	.4	$5\frac{1}{4}$	1.5	$5\frac{1}{2}$	$\frac{1}{8}$	$\frac{7}{16}$	$1\frac{1}{16}$	$7\frac{1}{8}$	8	$\frac{5}{8}$	$\frac{5}{16}$	1.3
$4\frac{1}{2}$	.244	$\frac{1}{4}$	$9\frac{1}{2}$	$5\frac{1}{16}$	6	1.37	$6\frac{5}{16}$	$\frac{5}{16}$	1.3	$\frac{3}{8}$	$1\frac{13}{16}$	7.25	$\frac{5}{8}$	.2.	4.	$5\frac{1}{8}$	1.75	$6\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{2}$	$1\frac{1}{16}$	$7\frac{1}{8}$	8	$\frac{3}{4}$	$\frac{5}{16}$	1.3
5	.256	$\frac{17}{64}$	$10\frac{1}{2}$	$5\frac{15}{16}$	$6\frac{1}{2}$	1.06	$7\frac{1}{8}$	$\frac{5}{16}$	1.2	$\frac{3}{8}$	$1\frac{7}{8}$	7	$\frac{5}{8}$	.21	.3	$6\frac{1}{8}$	1.5	$6\frac{3}{8}$	$\frac{3}{16}$	$\frac{1}{2}$	$1\frac{1}{16}$	$8\frac{1}{2}$	8	$\frac{3}{4}$	$\frac{5}{16}$	1.2
$5\frac{1}{2}$	.27	$\frac{17}{64}$	$10\frac{3}{4}$	$6\frac{3}{16}$	7	1.06	$7\frac{5}{8}$	$\frac{5}{16}$	1.2	$\frac{3}{8}$	$1\frac{15}{16}$	7.3	$\frac{5}{8}$	.23	.33	7	1.7	$7\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{2}$	$1\frac{1}{16}$	$9\frac{1}{8}$	8	$\frac{3}{4}$	$\frac{5}{16}$	1.2
6	.284	$\frac{9}{32}$	$11\frac{1}{4}$	$6\frac{7}{8}$	$7\frac{7}{8}$	1.2	$8\frac{1}{4}$	$\frac{5}{16}$	1.1	$\frac{3}{8}$	2	7.1	$\frac{5}{8}$	.22	.33	$7\frac{1}{2}$	1.6	$7\frac{7}{8}$	$\frac{3}{16}$	$\frac{5}{8}$	$1\frac{1}{16}$	$9\frac{5}{8}$	10	$\frac{3}{4}$	$\frac{3}{8}$	1.3
$6\frac{1}{2}$	.297	$\frac{19}{64}$	$11\frac{7}{8}$	$7\frac{5}{32}$	$8\frac{1}{8}$	1.	$8\frac{3}{4}$	$\frac{5}{16}$	1.08	$\frac{3}{8}$	$2\frac{1}{16}$	6.9	$\frac{11}{16}$	.23	.37	$8\frac{1}{8}$	1.6	$8\frac{1}{4}$	$\frac{3}{16}$	$\frac{5}{8}$	$1\frac{1}{16}$	$10\frac{1}{4}$	10	$\frac{3}{4}$	$\frac{3}{8}$	1.3
7	.31	$\frac{5}{16}$	$12\frac{1}{2}$	$7\frac{11}{16}$	$8\frac{3}{4}$	1.1	$9\frac{3}{8}$	$\frac{5}{16}$	1.	$\frac{3}{8}$	$2\frac{1}{8}$	6.8	$\frac{11}{16}$	.22	.37	$8\frac{5}{8}$	1.5	9	$\frac{3}{16}$	$\frac{5}{8}$	$1\frac{1}{4}$	$10\frac{7}{8}$	12	$\frac{3}{4}$	$\frac{3}{8}$	1.2
$7\frac{1}{2}$	.324	$\frac{21}{64}$	$13\frac{1}{2}$	$8\frac{7}{32}$	$9\frac{3}{8}$	1.14	10	$\frac{5}{16}$	.09	$\frac{3}{8}$	$2\frac{3}{8}$	6.6	$\frac{11}{16}$	.21	.37	$9\frac{1}{4}$	1.6	$9\frac{5}{8}$	$\frac{3}{16}$	$\frac{5}{8}$	$1\frac{1}{16}$	$11\frac{5}{8}$	12	$\frac{7}{8}$	$\frac{7}{16}$	1.3
8	.337	$\frac{11}{32}$	14	$8\frac{7}{32}$	$9\frac{7}{8}$	1.	$10\frac{3}{8}$	$\frac{5}{8}$	1.	$\frac{3}{8}$	$2\frac{1}{4}$	6.34	$\frac{3}{4}$	.22	.4.	$9\frac{7}{8}$	1.5	$10\frac{1}{8}$	$\frac{3}{16}$	$\frac{5}{8}$	$1\frac{1}{16}$	$12\frac{1}{8}$	12	$\frac{7}{8}$	$\frac{7}{16}$	1.3
$8\frac{1}{2}$	.35	$\frac{23}{64}$	$14\frac{5}{8}$	$9\frac{3}{32}$	$10\frac{1}{2}$	1.08	$11\frac{1}{4}$	$\frac{5}{8}$	.08	$\frac{3}{16}$	$2\frac{3}{8}$	6.6	$\frac{13}{16}$	.22	.43	$10\frac{3}{8}$	1.5	$10\frac{3}{4}$	$\frac{3}{16}$	$\frac{5}{8}$	$1\frac{3}{16}$	$12\frac{3}{4}$	12	$\frac{7}{8}$	$\frac{7}{16}$	1.2
9	.363	$\frac{3}{8}$	$15\frac{1}{4}$	$9\frac{19}{16}$	$11\frac{5}{8}$	1.08	$11\frac{7}{8}$	$\frac{5}{8}$	1.	$\frac{3}{16}$	$2\frac{7}{16}$	6.5	$\frac{13}{16}$	.22	.43	$10\frac{3}{8}$	1.4	$11\frac{1}{4}$	$\frac{3}{16}$	$\frac{5}{8}$	$1\frac{7}{8}$	$13\frac{1}{4}$	12	1	$\frac{1}{2}$	1.2
$9\frac{1}{2}$	.377	$\frac{3}{8}$	$15\frac{7}{8}$	$10\frac{5}{16}$	$11\frac{5}{8}$	1.08	$12\frac{3}{8}$	$\frac{5}{8}$	1.	$\frac{1}{4}$	$2\frac{1}{2}$	6.6	$\frac{13}{16}$	.22	.43	$11\frac{1}{2}$	1.6	$11\frac{7}{8}$	$\frac{3}{16}$	$\frac{5}{8}$	$1\frac{7}{8}$	$13\frac{7}{8}$	12	1	$\frac{1}{2}$	1.3
10	.39	$\frac{25}{64}$	$16\frac{1}{2}$	$10\frac{17}{32}$	$12\frac{1}{8}$	1.04	$12\frac{7}{8}$	$\frac{3}{8}$	.04	$\frac{1}{4}$	$2\frac{5}{8}$	6.6	$\frac{7}{8}$	.24	.35	12	1.8	$12\frac{1}{2}$	$\frac{1}{4}$	$\frac{5}{8}$	$1\frac{7}{8}$	$14\frac{1}{2}$	12	1	$\frac{1}{2}$	1.3
$10\frac{1}{2}$	.403	$\frac{13}{32}$	$17\frac{1}{8}$	$11\frac{5}{8}$	$12\frac{7}{8}$	1.08	$13\frac{1}{2}$	$\frac{3}{8}$	.03	$\frac{1}{4}$	$2\frac{5}{8}$	6.5	$\frac{7}{8}$	.21	.35	$12\frac{5}{8}$	1.5	$13\frac{1}{8}$	$\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{2}$	$15\frac{1}{8}$	14	1	$\frac{1}{2}$	1.2
11	.417	$\frac{27}{64}$	$17\frac{5}{8}$	$11\frac{29}{32}$	$13\frac{1}{4}$	1.	$14\frac{1}{8}$	$\frac{7}{16}$	.04	$\frac{1}{4}$	$2\frac{11}{16}$	6.4	$\frac{15}{16}$	.22	.38	$13\frac{1}{8}$	1.41	$13\frac{5}{8}$	$\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{2}$	$15\frac{5}{8}$	14	1	$\frac{1}{2}$	1.2
$11\frac{1}{2}$	.43	$\frac{7}{16}$	$18\frac{1}{4}$	$12\frac{7}{16}$	$13\frac{7}{8}$	1.	$14\frac{3}{4}$	$\frac{7}{16}$	1.	$\frac{1}{4}$	$2\frac{13}{16}$	6.4	1	.23	.4.	$13\frac{3}{4}$	1.5	$14\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{8}$	$1\frac{5}{8}$	$16\frac{1}{4}$	16	1	$\frac{9}{16}$	1.3
12	.443	$\frac{29}{64}$	$18\frac{3}{4}$	$12\frac{31}{32}$	$14\frac{3}{8}$	1.21	$15\frac{1}{4}$	$\frac{7}{16}$	.04	$\frac{1}{4}$	$2\frac{15}{16}$	6.4	1	.22	.4	$14\frac{1}{4}$	1.43	$14\frac{7}{8}$	$\frac{1}{4}$	$\frac{5}{8}$	$1\frac{5}{8}$	$16\frac{3}{4}$	16	1	$\frac{9}{16}$	1.3
13	.47	$\frac{15}{32}$	20	14	$15\frac{1}{2}$	1.	$16\frac{3}{8}$	$\frac{7}{16}$	.03	$\frac{1}{4}$	3	6.4	$1\frac{1}{16}$	.23	.42	$15\frac{3}{8}$	1.5	$15\frac{7}{8}$	$\frac{1}{4}$	$\frac{11}{16}$	$1\frac{5}{8}$	18	18	1	$\frac{9}{16}$	1.2
14	.496	$\frac{1}{2}$	$21\frac{1}{2}$	$15\frac{1}{16}$	$16\frac{5}{8}$	1.	$17\frac{1}{8}$	$\frac{1}{2}$	1.	$\frac{1}{4}$	$3\frac{1}{8}$	6.25	$1\frac{1}{8}$	.23	.45	$16\frac{1}{2}$	1.4	17	$\frac{1}{4}$	$\frac{3}{4}$	$1\frac{3}{4}$	$19\frac{1}{4}$	18	$1\frac{1}{8}$	$\frac{5}{8}$	1.3
15	.523	$\frac{33}{64}$	$22\frac{3}{4}$	$16\frac{1}{8}$	$17\frac{3}{4}$	1.	$18\frac{7}{8}$	$\frac{1}{2}$	.04	$\frac{1}{4}$	$3\frac{1}{4}$	6.1	$1\frac{1}{16}$	.22	.38	$17\frac{5}{8}$	1.4	$18\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{4}$	$1\frac{13}{16}$	$20\frac{1}{2}$	18	$1\frac{1}{8}$	$\frac{5}{8}$	1.2
16	.549	$\frac{35}{64}$	$23\frac{7}{8}$	$17\frac{29}{32}$	$18\frac{7}{8}$	1.	$19\frac{7}{8}$	$\frac{1}{2}$	.04	$\frac{1}{4}$	$3\frac{3}{8}$	6.2	$1\frac{1}{4}$	.23	.4	$18\frac{3}{4}$	1.5	$19\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{4}$	$1\frac{7}{8}$	$21\frac{3}{8}$	20	$1\frac{1}{8}$	$\frac{11}{16}$	1.3
17	.574	$\frac{37}{64}$	$25\frac{1}{2}$	$18\frac{31}{32}$	20	1.	21	$\frac{1}{2}$	.04	$\frac{1}{4}$	$3\frac{1}{2}$	6.	$1\frac{5}{16}$	.23	.42	$19\frac{7}{8}$	1.46	$20\frac{1}{2}$	$\frac{5}{16}$	$\frac{7}{8}$	$1\frac{15}{16}$	23	20	$1\frac{1}{4}$	$\frac{11}{16}$	1.2
18	.602	$\frac{39}{64}$	$24\frac{3}{4}$	$19\frac{25}{32}$	$21\frac{1}{8}$	1.	$22\frac{1}{8}$	$\frac{1}{2}$	.03	$\frac{1}{4}$	$3\frac{5}{8}$	6.	$1\frac{3}{8}$	.23	.37	21	1.4	$21\frac{3}{8}$	$\frac{5}{16}$	$\frac{7}{8}$	2	$24\frac{1}{2}$	22	$1\frac{1}{4}$	$\frac{3}{4}$	1.2
19	.629	$\frac{5}{8}$	$27\frac{1}{8}$	$20\frac{7}{8}$	$22\frac{1}{4}$	1.	$23\frac{1}{4}$	$\frac{1}{2}$	.8	$\frac{1}{4}$	$3\frac{3}{4}$	6.	$1\frac{7}{16}$	.23	.38	$22\frac{1}{8}$	1.45	$22\frac{7}{8}$	$\frac{5}{16}$	$\frac{7}{8}$	$2\frac{1}{16}$	$25\frac{5}{8}$	22	$1\frac{1}{4}$	$\frac{3}{4}$	1.2
20	.655	$\frac{41}{64}$	29	$21\frac{7}{8}$	$23\frac{3}{8}$	1.	$24\frac{3}{8}$	$\frac{1}{2}$	.8	$\frac{1}{4}$	$3\frac{7}{8}$	6.	$1\frac{1}{2}$	.23	.4	$23\frac{1}{4}$	1.4	24	$\frac{5}{16}$	$\frac{7}{8}$	$2\frac{1}{8}$	$26\frac{1}{2}$	24	$1\frac{1}{4}$	$\frac{13}{16}$	1.2

NOTE!!—PIPES SHOULD BE $T = \frac{P \times D}{10{,}000} + \frac{1}{16}$ " WHEN OV

AS THE WEIGHT BECOMES VERY EXCESSIVE O

VERSAL STANDARD.

RE

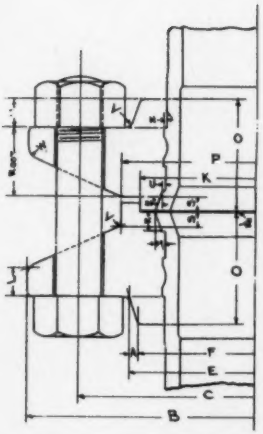
MAY BE OMITTED AS SHOWN BELOW.

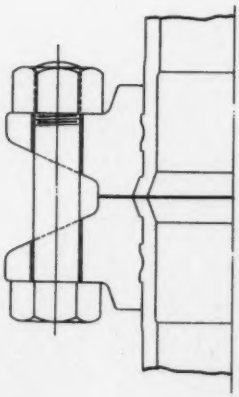
FIG. No. 8.

C. PITCH CIRCLE DIAM.	NUMBER OF BOLTS	DIAMETER OF STEEL BOLTS	R. DEPTH OF FLANGE BEYOND JOINT FACE	M.	U	N	V.	W.	BURSTING PRESSURE LBS. PER SQ. IN. ALLOWING 50000 LBS. TENSILE STRENGTH	LOAD ON EACH BOLT DUE TO BURSTING PRESSURE IN PIPE.	STRESS AT ROOT OF THREAD IN BOLTS DUE TO BURSTING PRESSURE OVER AREA OF PIPE TO INSIDE DIA. OF FLANGE.	STRESS ON BOLTS TO GIVE 10000 LBS PER SQ. IN. PRESSURE ON GASKET	STRESS IN FLANGE DUE TO MOMENT OF LOAD ON BOLT AT BURSTING PRESSURE OF PIPE.
2	4	5/8	1/4	1.3	1/8	5/32	1/8	3/16	9375	5850	48600	4570	18500
2 1/2	4	5/8	1/4	1.3	1/8	5/32	1/8	3/16	7500	7430	36800	3620	18500
3	4	5/8	1/4	1.2	1/8	5/32	3/16	3/16	6770	9750	48300	4595	18500
3 1/2	8	5/8	5/16	1.4	3/16	5/32	3/16	1/4	6250	8020	44200	4100	18200
4	8	5/8	5/16	1.3	3/16	5/32	3/16	1/4	5850	10700	53000	5000	20500
4 1/2	8	3/4	5/16	1.3	3/16	5/32	3/16	1/4	5550	12600	41800	4370	20700
5	8	3/4	5/16	1.2	3/16	5/32	3/16	1/4	5310	15100	58000	4560	24700
5 1/2	8	3/4	5/16	1.2	3/16	5/32	3/16	1/4	4830	16400	54200	4710	24400
6	10	3/4	5/16	1.3	1/4	5/32	3/16	1/4	4687	15000	49500	4550	21400
6 1/2	10	3/4	3/8	1.3	1/4	5/32	1/4	3/8	4560	16900	56000	4675	22400
7	12	3/4	3/8	1.2	5/16	5/32	1/4	3/8	4465	15900	52600	4800	23500
7 1/2	12	7/8	7/16	1.3	13/64	5/32	1/4	3/8	4375	17800	42400	4025	23600
8	12	7/8	7/16	1.3	1/32	1/16	1/4	3/8	4297	20300	48400	3935	25600
8 1/2	12	7/8	7/16	1.2	7/32	1/16	1/4	3/8	4225	22300	53300	4550	24400
9	12	1	1/2	1.3	1/4	1/16	1/4	3/8	4167	24600	44900	3990	22500
9 1/2	12	1	1/2	1.3	1/4	1/16	1/4	3/8	3950	25900	47100	4175	23200
10	12	1	1/2	1.3	1/4	1/16	1/4	3/8	3910	28200	51300	4220	23500
10 1/2	14	1	1/2	1.2	1/4	1/16	1/4	3/8	3875	26300	48000	4240	23400
11	14	1	1/2	1.2	1/4	1/16	5/16	7/16	3840	28400	51800	4420	24600
11 1/4	16	1	9/16	1.3	11/64	1/16	5/16	7/16	3800	26800	48900	4320	23500
11 1/2	16	1	9/16	1.3	11/64	1/16	5/16	7/16	3780	29100	53000	4500	22000
12	18	1	9/16	1.2	11/64	1/16	5/16	7/16	3600	28800	52400	4600	22300
12 1/4	18	1 1/8	5/8	1.3	13/64	1/16	5/16	1/2	3570	33300	48100	4050	22500
12 1/2	18	1 1/8	5/8	1.2	13/64	1/16	3/8	1/2	3540	37100	53600	4460	23300
13	20	1 1/8	11/16	1.3	15/64	1/16	3/8	1/2	3420	37100	53500	4525	22400
13	20	1 1/4	11/16	1.2	5/16	1/16	1/2	5/8	3400	40900	45800	3930	23800
13 1/4	22	1 1/4	3/4	1.2	21/64	1/16	1/2	5/8	3384	42000	47000	3970	24600
13 1/2	22	1 1/4	3/4	1.2	21/64	1/16	1/2	5/8	3290	45200	50400	4395	23000
14	24	1 1/4	13/16	1.2	23/64	1/16	1/2	5/8	3280	45700	51200	4250	22800

1/16" WHEN OVER 6" IN DIAMETER

EXCESSIVE OTHERWISE.





EXPERIENCES WITH NICLAUSSE BOILERS ON THE U. S. S. *NEVADA.**

By Lieutenant A. M. Cook, U. S. N., Senior Engineer Officer, Member.

My experiences with the Niclausse boilers on the U. S. S. *Nevada* have been so commonplace that it may seem uninteresting to the searcher after something new in marine water-tube boilers, yet I am tempted to recount some of these experiences.

The *Nevada* was placed in commission at Portsmouth, N. H., March 5, 1903, with C boiler in use for auxiliaries. I was informed by the contractor, the Bath Iron Works, that each boiler had, at that time, been in use about seventy days. Since commissioning, the four boilers have been in use for all purposes, to June 1, 1905, as follows:

Boiler A, 291 days, 19 hours.
B, 270 days, 14 hours.
C, 276 days, 19 hours.
D, 230 days, 1 hour.

On March 6, 1903, two days after commissioning, the draft failed, and the use of the steam lance for cleaning soot off the tubes gave only temporary relief; matters got worse until March 12th, when it became necessary to stop all auxiliaries except the feed pump in order to increase the steam pressure. On that day the forced-draft blowers were started and run for a short period, with the idea of blowing the soot off the tubes; and for several days thereafter no further trouble with the draft was experienced. It was found at that time that the draft would fail after any boiler had been in use for three or four days, and to avoid this it became routine to run both

* For a description of the U. S. S. *Nevada* see Vol. XV, page 105.

blowers at full speed for a short time each morning watch, using the lance about once a week. The soot settled on the horizontal baffles and the lance would not clear the heating surface; the use of the blowers, however, was effective and disposed of the soot by way of the smoke pipe.

In chronological order, the next thing that caused some anxiety was the attempt to remove the circulating tubes for examination, when it was found that the plugs had been set up too hard. It is quite probable that the mechanics responsible for this condition had had no other experience with the boilers than that of erecting them as new work. The plugs are made with a double thread on a taper, and after having been set up too hard, a little heat did the rest, and it was difficult to remove them; their removal was finally accomplished by patience, the use of extra tools and a wrench for clamping the generating tubes in their seats while unscrewing the plugs. In replacing these plugs it is necessary to coat the threads in with a lubricant which will not burn hard, to start the plugs in carefully, and then screw them home with a moderate strain. A 14-inch wrench with a strain of about 50 pounds is sufficient.

The next trouble was with grease in the boilers. When the ship was placed in commission there was no filtering material in the feed tanks. My attention being devoted to more pressing matters, this important requisite for the care of the boilers had been neglected. Loofa, or rag plant, has since been used with good results, and grease has caused no anxiety. During this experience without filtering material, it was found that the grease was confined almost entirely to the first three headers next the feed inlet, most of it being deposited on the inside of the circulating tubes of these headers. The feed water, in passing first through these headers, as it seems to do, deposited nearly all the grease. The tubes of the adjoining headers were found to have only a trace of grease. On the opposite side of the boiler, away from the feed inlet, no trace of grease was ever found, and, in consequence, it was necessary to examine and clean the tubes near the feed inlet oftener

than the others. The character of the circulation is, undoubtedly, the cause of finding all the grease in a few headers near the feed inlet and the salt scale in the headers farthest away. The water enters at one side of the boiler and apparently circulates through nearly all the headers in turn until it reaches the other side, at which point the concentration of the water is higher than anywhere else; therefore, the salinometer connection should be at the end of the drum farthest from the feed inlet, instead of in the center, as generally placed.

At the first general cleaning of the boilers, scale was found in all of them. Since then none has been found except in one or two instances, where the circulating tubes had become defective. The boilers had the same feed water, care and attention before as they did after this scale was found, and I am of the opinion that the scale was in the tubes when the ship was delivered to the Government. This opinion is strengthened by a later experience, when unusual and urgent circumstances, combined with a leak in the condenser, caused one boiler to be used for about fifteen hours, with a concentration equal to that of ordinary sea water, and an examination at the first opportunity showed no scale at all in the tubes. During the time that this boiler was in use with salt water, we were burning about fifteen pounds of coal per hour per square foot of grate, and there was not the slightest evidence of any priming. This was a revelation to me, for my torpedo-boat experience led me to believe that such a thing as no priming when using salt feed could not occur with any water-tube boilers. The scale found at the first general cleaning was removed with the turbine cleaner and scrapers, some of which were made on board, having two edges, shaped to the curve of the tube and held out by the spring of the fork on which they are formed. After the scale, grease, or any other kind of dirt, has been loosened in the tube it is removed by a steam jet. This jet is produced by a pipe, somewhat longer than the tube, connected with a steam hose. The end of this pipe is closed, and there are small holes near the end, drilled so as to direct the jet backwards at an angle of about 45 degrees. With the steam turned

on, this jet will remove all water and loose dirt from the tube. Most of the water in the large tubes is usually removed by a syphon before using the jet, on account of the large amount that would be blown into the fireroom.

After the ship had been cruising for some time in squadron, it was found that some of the malleable iron lanterns of the lower generating tubes were broken, as shown in Figure No. 1; the first one discovered to be in this condition had developed a slight leak around the outer cone joint; an occasional broken one was found in this way, but the greater number were not discovered until the circulating tubes were removed for the routine examination. Requisition was made for new lanterns. In the meanwhile boiler D was in use for some weeks, during which time it was known that several of the lanterns were broken. Some of these had leaked when they first broke, or, rather, when they first came to our notice; but when pressure was off the boiler, they were put straight on their seats and the dog adjusted reasonably tight against the lantern plug; this always stopped the leak. When the boiler was taken in hand for the renewal of these defective lanterns, it was found that not fewer than eleven were broken in the lowest row, out of a total of nineteen in that row. A few others broke in this boiler afterward, but it is probable that these had already been cracked. This trouble is believed to have been due to the material of which the lanterns were made and to the usage to which they were subjected; they are malleable castings, and the usage referred to was the practice of opening the fire doors to control the steam, which practice was of frequent occurrence while in squadron until the trouble of cracked lanterns was understood; since the opening of fire doors to control steam has been stopped, very few lanterns have broken. I believe that one of the old style lanterns is occasionally broken even now while cleaning fires. When the fire door is opened wide, with a hot fire, the great inrush of cold air striking the tubes of the bottom row contracts the lower side of them to a considerable extent, and very much more quickly than it does the top sides of the same tubes; this causes the tube to bend up in the

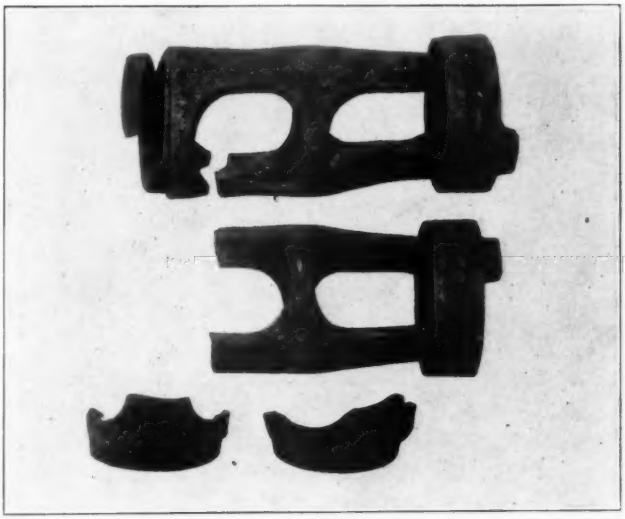


Fig. 1.—Broken Lanterns.

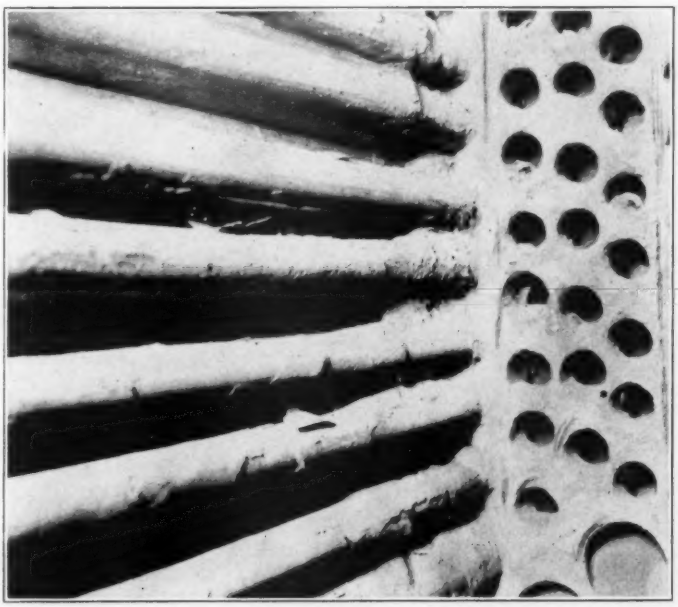


Fig. 2.—Tubes Before Cleaning.

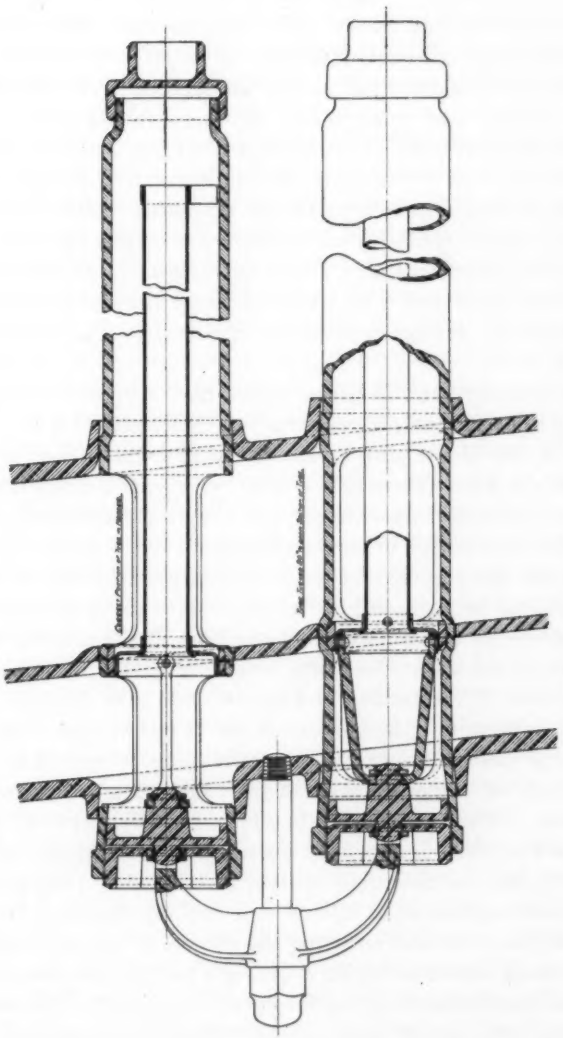


Fig. 3.—NEW TYPE OF LANTERNS.

middle in the form of an arch throughout its length; at the same time the lantern, being fast in the header, remains in its original position and causes what I am pleased to call a "kink" in the lantern. Either a frequent repetition of this action, or some especially violent one, causes the fracture. An inspection of the furnace not infrequently shows tubes bent up as described; in every case, however, these same tubes will be found to be straight at a later date, and I do not believe that this bending causes any other damage than the fracture of the lanterns. This trouble cannot occur with lanterns made according to present practice. The bending of tubes, however, shows the great strain to which straight tubes, fast to both ends, must be subjected in water-tube boilers with tubes so secured.

The manufacturers of these boilers have long since stopped making lanterns of the pattern originally supplied to this ship. A few of the new style ones have been in use in the *Nevada's* boilers for nearly a year with entire satisfaction. The new lantern is made in one piece, with the tube as shown on sectional elevation of the headers and tubes. (See Fig. 3.) The tubes are received by the manufacturer from the mill of one diameter throughout; the header ends of tubes are upset to compensate for the boring and threading on the inside; they are then expanded at the points where the joints are made on the fire side of the header, and where they pass through the header diaphragm; the loose ends are then swedged and the tubes are passed on to the machine shop, where they are machined, the lantern portion of the tube being milled out of the solid. The improved tubes referred to as having been used on the *Nevada* depended upon the fit of the plug thread for tightness; on the tubes shown, the threaded plug carries a swiveled cone ahead of it, this cone making the joint on the inside of the tube and relieving the thread of the necessity of being steam and water tight. This is a further improvement, and the substitution of a cone for a screw joint and should eliminate any sticking of these plugs. With this construction, which is but an extension of the cone principle for all joints

in this boiler, I am informed they will test tight at 500 pounds when screwed in with one hand and a 6-inch wrench. This construction does not eliminate the effect of expansion and contraction on the tubes, but its flexibility compensates for these strains, and, as stated, one year's service has not developed any fault.

In putting in new tubes it has happened that the tube did not fit properly in both joints of the header. A reamer is supplied with the boilers for the purpose of reaming out the seats in the header. This has not been used, the method followed being to lap off the cone of the tube which is too large. This latter method is used for two reasons: one, because it is the easiest way to correct the fault; the other, and the controlling reason, being that there is as much liability, if not more, that the fault is in the tube rather than in the header. If the header is changed, the next tube may not fit it, while if the tube is changed, it can always be put back in the same place, and whenever it becomes necessary to put in another tube the header will still be in its original condition.

I have gone into the troubles on board the *Nevada* quite extensively, believing this to be the best means of explaining the way in which they were overcome, and the necessity for precautions against their recurrence. The means taken may not have been the best in every case, but they have not been radically wrong. No boiler of this ship has ever had to be taken off at sea or in port on account of any defect. This is a very good showing for the design and workmanship.

A recital of the routine in caring for the boilers may now be of interest. The circulating tubes of the headers near the feed inlet and those of the lowest row of tubes across the boiler are removed for examination after the boiler has been in use from twelve hundred to fifteen hundred hours, these being cleaned at the same time, when necessary. A few other tubes, selected at random, are opened for inspection, a record being kept so that different ones will be taken each time. If there is any fault general in the whole boiler it will be developed by this means, while any purely local trouble might

be missed. To avoid this, all the circulating tubes were removed from all the boilers during the second year in commission. Two of the small tubes in the middle of boiler A were found to have defective circulating tubes; one of them had been partly filled with a deposit of scale and had to be renewed, the other was not injured sufficiently to require renewal.

I believe that it would be good policy to make a thorough inspection of all the tubes in a boiler at least once in each year, as while after the damage is done the cause will be found without this work, yet such inspection may prevent possible damage. This thorough examination of all the tubes in one of these boilers requires only a couple of days under ordinary conditions, and in a ship where all boilers are not constantly in use at sea the boilermaker can work at sea as well as in port, and there will be no trouble in finding the opportunity for this work.

When a boiler is to be out of use for more than a very few days it is always entirely filled with water. The water is tested for acidity, and enough alkali is added, if necessary, to be certain that the water is not acid.

Precautions were taken from the first to see that the gauge glasses were always in proper working order; they were blown through every watch. The water end of the glass is connected with the bottom of the drum by a U-shaped pipe, which is fitted with a cock at the bottom of the U. This is always blown before blowing the glass, and it is found that considerable sediment collects at this point. By this means the dirt is prevented from being raised into the smaller passages leading to the glass.

The boilers were originally installed with Niclausse automatic feed regulators. I have no doubt that these will work satisfactorily so long as they work at all, but they were never given a trial after the ship was put in commission. As soon as I took charge of the department, on March 5, 1903, they were all secured in the wide-open position; afterwards they were all removed. They are not necessary with these nor with any water-tube boiler in use in the U. S. Navy. The

regulation and control of the water level in the Niclausse boiler is not difficult in the least, under ordinary circumstances; one water tender having cared for four boilers when steaming at full power with forced draft. This has been done on the *Nevada* for four consecutive hours while burning 30 pounds of coal per hour per square foot of grate. I am firm in the belief that the use of automatic feed regulators is but an invitation for negligence and disaster. If they are in use, the water tender will naturally become negligent and not watch the water glass; when they cease to work is most probably the time when confidence in them will prevent the water tender from noticing that they are out of order. The water tender must be on duty, whether the automatic regulator is in use or not; if it would allow us to dispense with this valuable petty officer I might be converted to its use, but it does not do so.

There is always some grease in the feed water, and some dirt, to say nothing of salt, in spite of the filter and any other precautions that may be taken. It has been the routine to blow the boilers a short, hard blow twice daily; in the morning at 8 o'clock the surface blow is used, after being sure that the water in the boiler is at the level of a mark on the gauge such that the scum pan is just at the water level; in the evening, at 8 o'clock, the bottom blow is used, alternating on even and odd days with the blows on opposite sides of the boiler.

It had been found that the after boilers were less efficient than the forward boilers, and, in an effort to overcome this difference, if possible, it was decided to change the baffling in C boiler, one of the after ones, believing that the horizontal baffles were holding the soot above them in considerable quantity, and I thought that this accumulation would be reduced by removing an occasional baffle tube to allow the gases to go up through the baffling at intermediate points. To change the baffling it was necessary to remove a few generating tubes, and it was found that all the tubes thus removed were covered outside with a thick, hard scale, of a non-conducting material, and that

this scale was very hard to get off. Its removal required the use of steel scrapers and a considerable application of "elbow grease." This scale was from one-thirty-second to one-sixteenth of an inch in thickness. I believed that the cause of the lack of efficiency was located, and that it would be solved by removing all the tubes of this boiler and cleaning the fire side of them. Owing to the lack of causes requiring the removal of many tubes and the consequent inexperience of the boilermaker in that work, this was considered to be a more difficult task than it eventually proved to be. Advantage was taken of a stay in port to begin this work, and it was found that the conditions described were the same throughout that boiler. The work progressed very satisfactorily to completion. In view of the condition of C boiler, I determined to undertake this work on all boilers at the first opportunity. Boiler A, a forward one, was next taken in hand. It was at once seen that the condition of the fire side of the tubes of boiler A was entirely different from that which had existed in C boiler. There was a considerable amount of ordinary soot on the tubes and baffles, but it was of such character that a good blow from one's mouth was sufficient to remove it, leaving the tube perfectly clean. There was not a single tube in this boiler which had scale on it like that which had been found on every tube in C boiler. I am still looking for a satisfactory explanation of this peculiar condition found in boiler C.

Fig. 2 shows the condition usually found before cleaning the tubes. This dirt disappears entirely after blowing with the steam lance.

Boiler D was next examined. This is the other after boiler. Before taking up the work in A boiler, the tubes had not had the usual cleaning with the lance after the last steaming. Before starting D boiler I had one side cleaned in this way, leaving the other side in a condition supposed to approximate that which had existed in A boiler. It was found that the side not cleaned was very similar in condition and amount of dirt to what had been found in A boiler. There was no

hard scale such as had been found in C. On the side that had been cleaned with the lance the tubes were all clean and in perfect condition.

There was, however, an explanation for the fact that D boiler, notwithstanding its freedom of scale on tubes, had shown no greater efficiency than C. When the boilers were first installed, a different system of baffling had been put in. This had been changed before the trial trip, but many of the old baffles had not been removed from boiler D. It is believed that these extra baffles among the generating tubes of the boiler had caused the soot to collect very quickly after the boiler was started, and had thus had a bad effect on its efficiency.

To illustrate the facility with which the Niclausse boiler can be thoroughly overhauled, or re-tubed in emergency, I want to detail the work done on boiler D at Annapolis. The boiler was taken in hand at 10·00 A. M. Wednesday, May 24th, 1905; about four hundred and thirty circulating and generating tubes were taken out, cleaned where necessary both inside and outside, and all replaced; those not removed were cleaned in place; the boiler was ready to test at the end of working hours on Monday, May 29th. The test made on Tuesday morning, May 30th, showed that every tube was absolutely tight at 240 pounds pressure with cold water. The time taken for this work, not counting the filling of the boiler for testing, was about forty working hours. The test on A boiler after the work on it had been done was perfect. The test on C boiler, the first one undertaken, showed only one tube leaking slightly, and that was made tight in a few minutes. The men employed on this work were one boiler-maker and one water tender, with such help as could be used to advantage. About half the time the two men mentioned were alone on the job. The help was used principally for cleaning the tubes.

From the experience gained in this work I am free to say that in case of emergency, such as might occur in action, where a boiler had been wholly disabled, that the working force on

board could completely re-tube one boiler and have it in commission again in three days. This would allow time for renewing headers as well as tubes, in case any of the headers should have been damaged. It would not be necessary to call for any help from shore, or any special mechanics, or any tools other than those habitually in use with the boilers. The tubes can be removed in this boiler, cleaned and examined externally and internally, and the same tubes replaced, giving one the assurance that every tube in the boiler is properly cared for and sound. This, together with the comfortable feeling that one can depend on the continued service of the boilers, without the probability of failure when it is most important that they be kept going at full power, are strong and effective arguments in favor of this boiler.

I desire to say a word which has been repeated by every engineer who has had experience with Niclausse boilers and with whom I have talked, that is: "Before having had practical experience and been in contact with them, I looked questioningly at the multitude of cone joints. I am now convinced that they are one of the best points in the design of the boiler."

The brick work and casings of the boilers have not suffered to any appreciable extent. It has been necessary to rebuild portions of the side walls, but no metal work on sides or back has been necessary; the casing is as tight now as ever, and is not warped. The only trouble in this respect was with the door liners, and this was confined to the cast-iron arch. After a very little use with anything more than easy firing, these arches began to droop, interfering seriously with the firing. While at the Norfolk yard last spring the design of the arch was changed; this also involved some change in the lintel which helps to support the headers. The new work is protected to some extent by being hollow, with a provision for air circulation, and has stood well for a much longer period than in the case of the old castings.

In view of the fact that a recent inspection of the water side of these boilers showed the surface of the metal to be

absolutely free from pits, rust spots, or scale, and in perfect condition, a description of the method employed for testing water during the past two years might be of interest.

The litmus-paper test for acidity was found unsatisfactory, and at the suggestion of Mr. J. M. Cratty, chemist in the Navy Yard, Norfolk, phenolphthalein, a well known indicator in chemical analysis, was tried.

Half the amount that can be put on a ten cent piece, dissolved in 50 c.c. of alcohol, would be sufficient to last a ship for a year.

A drop of the solution in a test tube of the water to be tried will turn the whole red, if it is alkaline; but if the water is acid it will show no change in color.

The red color was generally found, but when this was not the case a small amount of soda added to the water in the boiler was sufficient to neutralize the acid.

In practice it would be advisable to have a solution of soda of such strength that a certain quantity, say one-half c.c., would have the same effect upon an ounce of water as a pound of soda would have on a thousand gallons. In this way the amount of soda required for any boiler could be readily determined.

A simple quantitative analysis for determining the chlorides in the boiler water has also been used. With a condenser ever so tight a small amount of salt will creep into the boilers after a time. When it is necessary to pump out a boiler for any purpose it is well to know if the water is good enough to use over, as feed water, or not. The hydrometer will not show this, but a small outfit with which to perform a simple chemical analysis will tell. Required are the following: A measure in ounces or c.c.; one burette, or dropper, graduated, preferably in c.c.; a standard solution of nitrate of silver; a solution of potassium chromate as an indicator; a white china dish or cup (an ordinary teacup will answer).

Have a solution of nitrate of silver of such strength that one c.c. of it will "kill" or react with one per cent. as much salt as is found in one ounce of ordinary sea water. As an illus-

tration, take one ounce of water from a boiler, place it in the cup, add a few drops of potassium chromate, which will give it a yellow color, and drop in slowly the standard solution of nitrate of silver. The water is to be kept stirred more or less according as the silver is dropped rapidly or slowly. A red spot will appear where the silver drops, but will at once dissipate until, say, 2½ c.c. of silver solution have been added, when the whole quantity of water will turn a dark red. The result of this test shows that there is 2½ per cent. as much salt in the water tested as would be in the same quantity of sea water. With a hydrometer this water would show no concentration. In the above case the water might well be used as feed water, but if the percentage were as much as four or five, it has been the custom to pump it overboard.

The outfit required for these two simple tests is not expensive, and could be readily supplied by the chemical laboratory at the Navy Yard. Such an outfit should be added to the allowance for steam engineering of every ship.

The standard solution of nitrate of silver can be made by dissolving 28 grams of crystals in one liter of pure water. The alcohol for dissolving the phenolphthalein must be neither acid nor alkaline for delicate tests. It will generally be found slightly acid, but this can be corrected by adding a small amount of caustic soda.

RECENT PERFORMANCE.

The above article was written in June of this year, and until August 14 there was no opportunity to test the effect of the work done in cleaning the fire side of the tubes. On that date a full-power trial of four hours duration was run. The coal used was good, but it is thought to have been no better than that used during a similar run in April. The following extracts from the record of these two runs show the results obtained:

	April 15.	August 14.
Collective I.H.P. for main and auxiliary machinery	2,476	2,767.6
Coal per hour, pounds	6,435	5,830
per square foot of grate	29.2	26.5
I.H.P.	2.60	2.10

The engines made an average of 194.7 revolutions for the four hours, working up from 140 revolutions at the time the record began. The last two hours the revolutions were 198 per minute. Much more steam could have been easily supplied by the boilers, and more could have been used by the engines, but the reducing valves in the main steam line would not permit of its passage. When this performance is compared with the trial record of the ship it will be seen that the boilers are still able to supply the demands of the engines, although they develop a great deal more power, owing to foul bottom and increased displacement.

U. S. S. *CHARLESTON.*

DESCRIPTION AND OFFICIAL TRIAL.

By Lieutenant M. E. Reed, U. S. Navy, Member.

The *Charleston* (Protected Cruiser No. 22), built by the Newport News Shipbuilding and Drydock Company, Newport News, Virginia, is one of the three twin screw, protected cruisers authorized by Congress in an Act approved June 7, 1900, and is the first of this class to have her trials. The other two sister vessels are the *St. Louis* and the *Milwaukee*, building, respectively, at Neafie & Levy Ship and Engine Building Company's Works, Philadelphia, Pa., and at Union Iron Works, San Francisco, Cal.

The contract for the *Charleston*, signed March 30, 1901, provided for delivery of the completed vessel within thirty-six (36) months from that date; that is, on or before March 30, 1904. This time limit was extended by the Department, however, first on November 11, 1903, for a period of twelve (12) months, to March 30, 1905; second on September 1, 1904, for a period of two (2) months and fifteen (15) days, to June 9, 1905, the time from the expiration of this last extension to the final delivery of the vessel to the Government being still subject to adjustment by the Department. With these settled extensions a total allowed time before delivery accrued amounting to fifty (50) months and fifteen (15) days. The usual time penalties for failure to complete the work within the contract time are stipulated in the contract.

The total price to be paid for the vessel, exclusive of certain portions of the armor, armament, etc., which are supplied by the Government, is two million seven hundred and forty thousand dollars ($2,740,000), of which one million two hundred and forty thousand dollars ($1,240,000) was allotted for

the building and complete installation of the machinery in readiness for service.

The guaranteed speed of the *Charleston* is 22 knots per hour maintained for four consecutive hours, on a mean draught of 22 feet 6 inches, corresponding to a displacement of about 9,700 tons. The designed power to produce this speed is about 21,000 collective indicated horsepower of propelling machinery at about 133 revolutions per minute of the main engines. Failure to reach the guaranteed speed of 22 knots under the stipulated trial conditions entailed penalties at the rate of fifty thousand dollars ($50,000) per quarter knot for speeds between 22 knots and 21½ knots, and at the rate of one hundred thousand dollars ($100,000) per quarter knot for speeds between 21½ knots and 21 knots. In case of failure to reach a speed of 21 knots it was optional with the Secretary of the Navy to reject the vessel or to accept her at a reduced price and on conditions to be agreed upon.

The keel of the *Charleston* was laid January 30, 1902, ten months after the signing of the contract; the hull was launched January 23, 1904, two years nine months and twenty-three days after the signing of the contract and one year eleven months and twenty-three days after the laying of the first keel plate.

HULL DATA.

Length between perpendiculars, feet	424
on load water line, feet	424
over all, feet and inches	426- 4½
of straight keel, feet and inches	352- 9
Beam, extreme, feet	66
at L.W.L. (22 feet 6 inches), feet	66
molded, feet and inches	65- 1
Ratio of length to beam	6.4599
Depth, molded, main deck side at M. S., feet and inches	40- 0½
upper deck side at M. S., feet and inches	47- 6½
Draught, normal, mean, feet and inches	22- 6
Displacement, mean load (empty bunkers), tons	9,148.2
normal (22 feet 6 inches), tons*	9,700
full load (24 feet 7 inches), tons	10,839
per inch at L.W.L. (22 feet 6 inches), tons	44.6

* 650 tons coal, 50 tons reserve feed water.

Area of midship section to L.W.L., square feet	1,312.5
L.W.L. plane (22 feet 6 inches), square feet	18,730
Wetted surface, square feet	31,850
Center of gravity of L.W.L. plane aft of M. S., feet	2.45
above bottom of keel, feet	22.5
buoyancy above bottom of keel, feet and inches	12- 8¾
aft of M. S., feet	68
Transverse metacenter above center of buoyancy, feet and inches,	12- 8¾
Longitudinal metacenter above center of buoyancy, ft. and ins.	507- 6
Coefficient of fineness, block	.536
of M. S	.884
of L.W.L	.665
cylindrical	.606
Number of frames	113
watertight compartments	183

Masts and Spars.—There are two steel military masts, the foremast located at frame 31, and the mainmast at frame 80, each fitted with a lower and an upper military top, a searchlight platform, spar topmast and signal yard. On the foremast, above the signal yard, is a lookout station fitted with guard rails and having speaking-tube connections to the deck below. On the mainmast is a small gaff for ensign hoist, etc., and this mast is also fitted for wireless telegraphy.

Decks and Bridges.—The *flying bridge*, 60 feet in length, extends across the ship at frames 32–33–34, the wings being six feet in width. Around the foremast is an extension on which are located steering wheel, helm and revolution indicators, engine telegraphs, whistle and siren pulls, voice-tube stand, all necessary means of communication and instruments for the purposes of navigation. A searchlight is located on each wing of this bridge, with control stands for electric night signals and for truck lights on the port and starboard wings, respectively. Near the mast is a small ammunition winch for hoisting to the tops.

The *forward bridge* is in the form of a triangle, the forward angle being rounded off, corresponding in plan to the forward part of the upper deck. This bridge is arranged for use in bad weather, with a protected conning and steering station having a curved front with sliding windows and doorless openings at the sides. This station encloses a set of naviga-

tional accessories similar to those on the flying bridge. On this bridge is the chart house, fitted for the use of the navigator, with chart tables, cases for charts, instruments, etc., and having tightly-shuttered sliding windows arranged for screening lights during night work. Cage mounts are provided on this bridge for two .30-caliber automatic and two .30-caliber machine guns.

The *after bridge*, immediately above and corresponding in plan to the after end of the upper deck, is fitted for signaling and for wireless telegraph operations. There is a house amidships for this latter purpose, on the top of which are also mounted two searchlights. Four automatic .30-caliber guns are mounted on this bridge.

HEIGHTS ABOVE L.W.L. (22 FEET 6 INCHES FORWARD AND AFT).

(Designed 22 feet 6 inches W.L. is 22 feet 4 1/8 inches above base line.)

Masts, fore and main, top of truck light, feet and inches	108- 8 1/8
Rangeing platform, floor ("crow's nest," forward), feet and inches	90-10 3/8
Upper military top, floor, both,* feet and inches	67- 0 3/8
Lower military top, floor, both,* feet and inches	56- 0 3/8
Searchlight platforms, both, feet and inches	75- 0 3/8
Flying bridge at C. L., frame 34, feet and inches	41- 9 3/8
at side, frame 34, feet and inches	41- 2 3/8
above main deck, feet and inches	23- 3
Forward bridge at side, frame 34, feet and inches	34- 2 3/8
After bridge at side, frame 80, feet and inches	34- 3 3/8
Upper deck at side, M. S., frame 57, feet and inches	25- 1 1/8
Flat armor deck, top, feet and inches	3- 5
Main deck planking at stem, top, feet and inches	21- 3 5/8
stern, top, feet and inches	20- 3 1/8
Waist plate at stem, top, feet and inches	24-11 1/8
Beading at stern, top, feet and inches	20- 9 3/8
Wireless telegraph station, top, feet and inches	42- 7 1/8
Smoke pipes, top, feet and inches	77- 7 3/8
Axes of 6-inch guns, gun deck, feet and inches	14- 6 3/8
main deck, feet and inches	21- 9 3/8

The *upper deck* extends from frame 28 forward to frame 87 aft, being cut away at the corners to give greater arc of fire to the 6-inch guns forward and aft, and at the sides near the cor-

* Depth of military tops, 2 feet 6 inches.

ners to give direct ahead and astern fire, respectively, to the two pairs of main-deck 6-inch guns forward and aft. Except under the bridges, where hammock nettings are fitted, the rail of this deck is clear, and is fitted only with stanchions and life lines. A day cabin for the commanding officer is built on this deck under the forward bridge, corresponding in size to the chart house above it, and fitted with bunk, settee, lockers and toilet stand. The boats, except two life boats, are stowed on this deck when at sea, in chocks and cradles at the eight positions convenient for handling by the four boat cranes when swung to their extreme forward and extreme aft train.

Type.		Location.	Handled by —— crane in —— position.	Weight, pounds, in service, with crew.
Steam cutter,	36-foot.	S.	10-ton, S., aft.	16,961
Steam cutter,	33-foot.	P.	10-ton, P., forward.	14,436
Sailing launch,	36-foot.	P.	10-ton, P., aft.	9,954
Sailing launch,	33-foot.	S.	10-ton, S., forward.	8,471
Cutter,	30-foot.	S.	3-ton, S., forward.	5,625
Cutter,	30-foot.	P.	3-ton, P., forward.	5,625
Cutter,	26-foot.	S.	3-ton, S., forward.	4,447
Cutter,	26-foot.	P.	3-ton, P., forward.	4,447
Cutter,	26-foot.	S.	3-ton, S., aft.	4,447
Cutter,	26-foot.	P.	3-ton, P., aft.	4,447
Cutter,	26-foot.	S.	10-ton, S., forward.	4,447
Cutter,	26-foot.	P.	10-ton, P., aft.	4,447
Whaleboat,	30-foot.	S.	On sea davits.	4,802
Whaleboat,	30-foot.	P.	On sea davits.	4,802
Whaleboat (gig),	30-foot.	S.	3-ton, S., aft.	4,471
Dinghy,	20-foot.	S.	3-ton, S., aft.	2,080
Dinghy,	20-foot.	P.	3-ton, P., aft.	2,080
Dinghy,	14-foot.	S.	3-ton, S., aft.	1,325
Punt,	14-foot.	S.	3-ton, S., forward.	700 (about).
Punt,	14-foot.	P.	3-ton, P., forward.	700 (about).

The engine-room and fireroom hatch enclosures and all other trunks or companion ways end at the upper deck with suitable coamings, gratings and skylights or canopies.

In connection with the lower compartments there are on this deck: eight fireroom ventilators, in pairs, port and starboard, at frames 39, 48, 56, 65; four engine-room ventilators, in pairs, port and starboard, at frame spaces 72–73, 81–82; two galley ventilators, port and starboard, at frame space 53–54;

and three small ventilators for air supply to small electric blowers. All ventilators for natural ventilation are fitted with rotating cowl heads.

Four smoke pipes, each connected with four main boilers, rise 100 feet from the base line of the ship and about 52 feet 6 inches from the upper deck. The outside diameter of smoke pipes is 8 feet 6 inches.

Four electrically-operated boat cranes are placed on this deck, their heels being stepped on the second (gun) deck below. The forward pair, at frame space 47–48, port and starboard, are designed for 3 tons dead-lift capacity; the after pair, at frame space 66–67, port and starboard, being designed for 10 tons dead-lift capacity. The boat-hoist duty of these cranes has been shown in boat table, and their motors will be referred to elsewhere.

Four electrically-operated deck winches for coaling and general duty are located on this deck, in pairs, port and starboard, at frame space 44–45 and 67–68.

Two electrically-operated ammunition-hoist winches for hoisting to the tops are located, one below the after side of the forward bridge and one below the forward side of the after bridge.

Twelve 3-inch R. F. guns are mounted on this deck, in pairs, port and starboard, at frames 28, 31, 45, 69, 82, 86, on regulation cage mounts; two 1-pounder R. F. guns, port and starboard, at frame 36; and two automatic .30-caliber guns, port and starboard, at frame space 78–79.

A horizontal, cylindrical fresh-water tank of 2,000 gallons capacity is located on this deck, having a dividing diaphragm fitted at its midlength, separating the tank into two equal compartments, port and starboard, the axis of the tank being athwartships. This tank is piped for gravity supply to the galley, bakery and those lavatories and pantries below which are not supplied by separate tanks, and is fitted with locked manhole plates, water glasses and overflow vents.

The *main deck* is the first of the continuous decks extending the whole length of the ship. Forward on this deck are the four deck openings of the hawse pipes for bower and sheet

chains, two riding bitts for anchor chains, a double-headed towing bitt amidships, four chain controllers with lever throw-outs, two anchor davits for handling the Navy-type anchors to the bill boards, bow stoppers, ring bolts, etc.

In a covered pocket between the main and gun decks, and rising above the main deck to a level below the axis of the forward 6-inch gun, is located the anchor windlass, the chains passing to the wildcats through chain pipes in the forward side of this pocket.

Leadsman's platforms are located outside the rail at frame 17, port and starboard.

Two electrically-operated deck winches for anchor work, boat hoisting, coaling and general duty are located at frame 26, port and starboard.

Within the superstructure on the main deck is berthing space for the crew, hammock nettings being fitted throughout at sides where not in wake of battery, doors, etc.

Aside from the trunks for ventilation, communication and access to lower compartments there are on this deck the following:

General mess issuing room, around foremast;

Bakery, forward of second smoke pipe, amidships;

Galleys, officers' and crew's, between second and third smoke pipes;

Wash-deck gear room, port side, opposite fourth smoke pipe;

Bread room, starboard side, opposite fourth smoke pipe;

Paymaster's office, forward side engine-room hatch;

Battalion lockers, after side engine-room hatch;

Executive officer's office, after compartment, starboard side;

Navigator's office, after compartment, port side.

Aft, on the main deck, outside the superstructure, are the usual companionways, skylights and ventilating trunks; also one electrically-operated general-service winch at frame 94, amidships.

Suitable warping and towing chocks, and double-headed

warping and towing bitts are distributed in convenient positions, forward and aft, for warping and towing.

The battery on this deck, six 6-inch guns and six 3-inch guns, is arranged as described under *Armament.*

Boat davits are fitted for two 30-foot cutters and for two 20-foot dinghies abreast of forward end of superstructure, and for two 30-foot whale boats abreast of after end of superstructure, all in pairs, port and starboard.

The *gun deck* is arranged as follows from forward :

Crew's shower baths ;
Crew's water closets ;
Windlass engine and windlass compartment (pocket) ;
Petty officers' and machinists' water closets ;
Berthing spaces for crew ;
Commissary's issuing room, port ;
Paymaster's issuing room, port ;
Laundry, starboard ;
Upper casemate, armored ;
Armory, forward side engine-room hatch ;
Officers' lavatories, port ;
Wardroom officers' rooms, four starboard, two port ;
Ordnance officer's office, starboard ;
Wardroom mess room, athwartships ;
Captain's office, pantry, cabins (main and after), staterooms (two) and baths (two).

On this deck are eight 6-inch guns, twelve 3-inch guns and two 1-pounder R.F. guns arranged as described under *Armament.*

The *berth deck* is arranged as follows:

Lamp room, paint room (port), oil room (starboard), prisons (two), all in forward compartment ;

Master-at-arms' room (port), band room (starboard), sergeant of marines' room (port), isolation ward (starboard), crew's berthing space, all in second compartment ;

Crew's berthing space, chain lockers, in third compartment ;

Sick bay, sick-bay bath room (port), dispensary (starboard), operating room (starboard), all in fourth compartment ;

Chief machinists' quarters and lavatory (port), chief petty officers' quarters and lavatory (starboard), crew's berthing space, all in fifth compartment;

Coal bunkers B–101, B–105, B–111, B–115 (starboard), B–102, B–106, B–110, B–114 (port), blacksmith shop (forward end of first uptakes), firemen's wash rooms (three, amidships, forward and after end of second uptakes and forward end of third uptakes), evaporator room, distillers in trunk above (between third and fourth uptakes), servants' wash room (after end of fourth uptakes), general work shop (port), refrigerating rooms (starboard), engineer's office (central, forward end of engine-room hatch), ice-machine room (starboard), ordnance, navigator's, warrant officers' storerooms (three, port), wardroom, marines', junior officers' storerooms (three, starboard), warrant officers' pantry (port), junior officers' pantry (starboard), all within the lower armored casemate, sixth compartment.

Warrant officers' quarters, mess room and five rooms (port), and one additional room (starboard), junior officers' quarters, mess room and four rooms (starboard), all in seventh compartment;

Wardroom rooms (six port and six starboard), in eighth compartment;

Captain's storerooms (two, port and starboard), blower room (right aft), in ninth compartment.

Electrically-operated ventilating blowers are located on this deck, as described under *Ventilation.*

The *protective deck* extends throughout the length of the vessel, being flat amidships and sloping downward to the sides and ends below the water line. The deck level corresponds with that of the berth deck, the spaces between the two decks at the sides and ends of the vessel being utilized for stores appropriate to the berth-deck spaces over them. The coal bunkers mentioned under the heading *berth deck* extend to the protective deck and are in communication with the lower bunkers under them by suitable doors in the protective deck.

The *upper platforms* are directly below the protective deck and are arranged for stores, magazines, ammunition passages, steering gear, etc. From frame 7, forward, these compartments are, in order, as follows:

General stores;
Bread room;
Ordnance storeroom (port);
Commissary's storeroom (port);
Paymaster's storerooms (two, port);
6-inch magazines (two, port and starboard);
Communicating room (central);
6-inch shell rooms (two, port and starboard);
Upper dynamo room (central);
Ammunition passages (two, port and starboard through boiler spaces, then one, central, through engine space);
6-inch shell room (port);
6-inch magazine (starboard);
6-inch magazine (athwartships);
Steering-engine room;
Steering-gear room.

The *lower platforms* are next below the upper platforms and are also arranged for stores, magazines, etc. Between frames 13 and 30, forward, are:

Construction storerooms (two port and three starboard);
Equipment storeroom (port);
6-inch magazine (athwartships);
3-inch magazine (port);
3-pounder and 1-pounder ammunition room (starboard);
Small arms ammunition room (starboard);

Between frames 83½ and 99, aft, are:

Engineers' storerooms (two, port and starboard);
Small arms ammunition rooms (two, port and central);
3-pounder and 1-pounder ammunition room (port);
6-inch magazine (starboard);
Dry-provisions rooms (two port, three starboard);
Saluting-powder room (starboard);
Sail room (port).

The *hold* is arranged as follows:

Trimming tanks (two, first and second compartments);
Store space, including fresh water tanks (third compartment);
3-inch magazine (fourth compartment);
Electrical storerooms (two, port and starboard);
Lower dynamo room (central);
Boiler rooms (four central, across center line);
Coal bunkers B–5, B–7, B–9, B–11 (starboard), B–6, B–8, B–10, B–12 (port);
Engine rooms (two, port and starboard);
Shaft alleys (two, port and starboard);
Engineers' storerooms (two, port and starboard);
3-inch magazine (central);
Water tanks (athwartships);
Trimming tanks (two, after compartments).

The *double-bottom compartments* are described hereafter under watertight compartments.

Framing.—All hull material not otherwise specified will be of mild steel of domestic manufacture.

The longitudinal framing consists of a vertical keel, five longitudinals on each side of this keel and below the protective deck, one stringer each side behind the armor about midway between protective and berth deck, and bilge keels at the fourth longitudinals (port and starboard). The *vertical keel* is 39 inches in depth except forward of frame 7 and abaft frame 102, where it tapers to its connection with the stem and stern post, and is continuous throughout. The plating of this keel is 20 pounds per square foot from frame 23 to frame 84 and 17½ pounds forward of frame 23 and aft of frame 84. The keel is not made watertight except within the space devoted to reserve fresh water where the compartments are divided, port and starboard, all other midship compartments extending across the center line so far as watertightness is concerned. Lightening holes are cut in the vertical keel forward and abaft the limits of the double bottoms. The *longitudinals* are continuous throughout the double bottoms and are carried as far to the ends of the vessel as consistent with

good work. Within the limits of the double bottoms they are of 17½-pound plate, and beyond this limit they are of 15-pound plate. The third longitudinal only, port and starboard, is made watertight, the others being pierced with lightening holes, drainage holes being provided to allow water to flow to the lowest pockets of compartments. The *stringer* behind armor is worked with 15-pound plates between frames and with a continuous Z-bar 6 inches by 3 inches by 3 inches across the inside of the frames. It extends forward to the stem, where it forms a breast hook, and aft as far as possible. The *bilge keels* are composed of two 15-pound plates about 24 inches in width forming a sharp V section, the inner space of which is filled in with yellow pine, and the outer edges of which are finished and stiffened by 2½-inch half-round bars, one on each side of the seam. These keels extend about 200 feet amidships and are secured to the outside plating over the fourth longitudinal. The *stem* is of cast steel, in two pieces, scarfed at about the normal load line and fitted to receive the outside plating, keel plates, deck and breast-hook connections. The *sternpost* is also of cast steel, fitted to receive plating, keel plates, etc., and for carrying the rudder. Provision is made for tiller-post stuffing box, packing and gland, hardover stops for rudder, and bushed lug for rudder pintle.

The *transverse frames*, 113 in number, are spaced 3 feet 6 inches apart from the stem to frame 35 and from frame 84 to the stern. Between frames 35 and 84, the boundaries of the boiler and machinery spaces, the frames are spaced 4 feet apart. In all cases intermediate frames are worked where required for additional strength or stiffness. The framing of the superstructure above the line of the main deck is not continuous with the structure below, but is secured to the main-deck plating by bracket plates.

Plating.—In general, all *outside bottom plating*, below the line of the protective deck, is raised and sunken, with lapped butts, except where doubled. This plating is of 23 pounds per square foot for a distance of about 200 feet amidships and is reduced to 20 pounds at the ends in two reductions. The

outside topside plating, above the line of the protective deck, is worked flush and is of 30 pounds per square foot behind armor diminishing to 20 pounds at the ends in five reductions. The *flat keel plates* are worked flush in two thicknesses, the inner plate of 23 pounds per square foot reduced at the forward end to 20 pounds, and the outer plate of 25 pounds per square foot reduced at the forward end to 22 pounds, the reduction of these plates forward being made in two steps.

Watertight Compartments.—There are 183 watertight compartments, of which 120 are below and 63 are above the protective deck.

Of those below the protective deck B–1, B–2, B–3, B–4 are boiler compartments; C–1, C–2 are engine compartments; D–2, D–3 are shaft-alley compartments; D–4, D–5, D–12, D–13 are engineer storerooms; B–73 to B–99, inclusive, are double-bottom compartments under boiler rooms (of which B–73 to B–88, inclusive, are wing compartments under coal bunkers and outboard sides of boiler rooms); B–89 to B–93, inclusive, are central compartments extending between the third longitudinals, port and starboard, and non-watertight at center line; B–94, B–95, B–96, B–97, B–98, B–99 are alternate port and starboard compartments fitted for *reserve feedwater tanks*, the entrance manholes being in the after two firerooms and of the high-coaming type); C–91 to C–99, inclusive, are double-bottom compartments under engine rooms; B–5, B–6, B–7, B–8, B–9, B–10, B–11, B–12 are lower coal bunkers abreast of boiler rooms.

Of those above the protective deck B–100 contains the hatch enclosures and passages for the forward two boiler rooms; B–109 contains the hatch enclosures and passages for the after two boiler rooms and includes the evaporator room; B–101, B–102, B–105, B–106, B–110, B–111, B–114, B–115 are upper coal bunkers over boiler rooms; C–106 contains the engine-hatch enclosure; that portion of C–100 below the berth-deck level between frames 68 and 79 at port side is assigned as additional space for engineering stores and is reached through scuttles in the machine shop and ordnance storeroom.

COAL-BUNKER CAPACITIES.

(Measured to 6 inches below bulb of deck beams, at 43 cubic feet per ton.)

Bunkers in hold.	Cubic feet.	Tons.	
B- 5	3,417	79.5	
B- 6	3,417	79.5	
B- 7	4,795	111.5	
B- 8	4,795	111.5	
B- 9	5,296	123.2	
B-10	5,296	123.2	
B-11	5,427	126.2	
B-12	5,427	126.2	
			880.8
Bunkers above protective deck.			
B-101	4,426	102.9	
B-102	4,426	102.9	
B-105	4,306	100.1	
B-106	4,306	100.1	
B-110	4,399	102.3	
B-111	4,399	102.3	
B-114	4,285	99.7	
B-115	4,285	99.7	
			810
Total,	72,702		1,690.8

DOUBLE-BOTTOM AND RESERVE-FEED WATER-TANK CAPACITIES.

Compartment.	Gallons.	Salt water, tons.	Fresh water, tons.
A-97	7,622.12	29.11	28.29
A-98	18,535.34	70.79	68.8
A-99	11,738.79	53.86	52.46
B-89	14,373.8	65.95	64.24
B-90	11,751.86	53.92	52.52
B-91	11,950.2	54.83	53.4
B-92	12,122.38	55.62	54.17
B-93	12,205.2	56	54.54

Compartment.	Gallons.	Salt water, tons.	Fresh water, tons.
B–94 (Reserve feed)	6,115.68	28.06	27.33
B–95 (Reserve feed)	6,115.68	28.06	27.33
B–96 (Reserve feed)	6,085.16	27.92	27.19
B–97 (Reserve feed)	6,085.16	27.92	27.19
B–98 (Reserve feed)	7,519.28	34.5	33.6
B–99 (Reserve feed)	7,519.28	34.5	33.6
	39,440.24	180.96	176.24
C–97	14,347.65	65.83	64.12
C–98	13,506.36	61.97	60.36
C–99	12,811.1	58.78	57.25
D–98	9,372.44	35.8	34.8
D–99	6,359.96	25.06	24.36
Total,	196,137.54	868.48	845.56

TRIMMING-TANK CAPACITIES.

Compartment.	Gallons.	Salt water, tons.	Fresh water, tons.
A– 1, forward,	9,910.19	45.47	44.29
A– 2, forward,	14,402.14	66.08	64.36
D– 9, aft,	10,389.68	47.67	46.43
D–10, aft,	12,011.25	55.11	53.68
Total,	46,713.26	214.33	208.76

FRESH-WATER TANK CAPACITIES.

Compartment.	Tank.	Gallons.	Tons.
A–3	13–15, Port,	816	3.04
A–3	13–15, Starboard,	816	3.04
A–3	16–18, Port,	1,484	5.53
A–3	16–18, Starboard,	1,484	5.53
A–3	19–20, Port,	1,535	5.73
A–3	19–20, Starboard,	1,535	5.73
D–8	Port,	1,182	4.41
D–8	Starboard,	1,182	4.41
Upper deck,	Port gravity,	1,000	3.71
Upper deck,	Starboard gravity,	1,000	3.71
	Total,	12,034	44.84

Coffer Dams.—A continuous coffer-dam belt about 30 inches wide extends the entire length of the ship at sides, from frame 7 forward to frame 109 aft, occupying the space on the slope of the protective deck outboard and rising to the level of the berth deck. The compartments comprised in this belt are: A–101, –104, –105, –108, –109, –112, –113, –116, –117; B–103, –104, –107, –108, –112, –113, –116, –117; C–102, –103, –104, –105; D–102, –103, –106, –107, –109, –110.

After the preliminary trials of the vessel these coffer dams are to be packed with suitable fire-proofed, corn-pith cellulose having the compressed surfaces laid in the horizontal plane.

Armor and Protection.—On the 20-pound plating of the *protective deck* are worked nickel-steel plates 1½ inches thick on the flats and 2½ inches thick on the slopes, giving a total thickness of 2 inches on the flats and 3 inches on the slopes. All hatches and openings through the protective deck are fitted with shutters of the same thickness as the deck, or with armor bars of approved strength.

The *main-belt side armor* extends between frames 35 and 84, a distance of 196 feet, abreast of the boiler and engine compartments. This belt is 4 inches in thickness and 7 feet 6 inches in height, of which 3 feet 5 inches is above the 22 feet 6 inches load water line. The upper edge of this belt reaches to the berth-deck level, the athwartships triangular spaces between the protective deck and berth deck at the ends of the side armor being filled in with 3-inch armor.

The *lower casemate*, extending from berth deck to gun-deck level, corresponds in thickness and area to the main side-armor belt, the side armor being 4 inches thick, and the athwartships armor at ends being 3 inches thick.

The *upper casemate* extends from gun deck to main-deck level, and, like the lower casemate, is composed of 4-inch side armor and 3-inch athwartship armor at ends. The length of this casemate is only 128 feet, however, as compared with the 196 feet of the lower casemate and main belt, extending only between frames 41 and 73 with an extension of the side armor at the corners over the next frame space forward and aft,

making the sides 136 feet in length. The after athwartships armor of this casemate coincides with the forward end of engine-room hatch at that level, while the forward athwartships armor coincides with the after side of the uptake enclosure for the forward group of boilers at that level. The upper casemate encloses the eight 6-inch gun positions on the gun deck, the gun positions being separated by 2-inch nickel-steel splinter bulkheads extending inboard about 15 feet from the side.

An armored *conning tower* is located at frame 28. It is elliptical in shape with its long axis athwartships, and has its entrance at the rear protected by a concentric shield plate which gives an 18-inch access passage, having the same curve as the tower. The height of the conning tower is 7 feet 6 inches and its axes are 6 feet and 8 feet, respectively. The thickness of both tower and shield plate is 5 inches. The top of the conning tower is covered by a 3-inch plate and of the access passage by a 3-inch sloped plate, the bottom plate, 2 inches thick, being shaped to include the access passage and to support the shield plate.

An *armored tube*, 30 inches inside diameter and 3 inches thick, is fitted from the floor of the conning tower to the protective deck. It provides for the leads of voice pipes, wiring, etc., for the various conning-tower devices and also affords access to the conning tower from the communicating room, ladder rungs being fitted inside the tube for this purpose.

Four *sponsons* of 4-inch armor give local protection to the four 6-inch gun positions at the corners of the superstructure on the main deck. The armor extends from the top of the upper casemate to the upper-deck level and is curved inward around the gun positions to afford protection in a fore-and-aft direction.

The twelve 3-inch gun positions on the gun deck outside of the upper casemate are protected by 3-inch nickel-steel *port plates* about 18 feet 6 inches long, extending over the space between the gun and upper decks. The forward and after 6-inch gun positions, on main deck outside of superstructure,

and the remainder of the secondary-battery positions not previously mentioned, are not provided with armor protection.

Armament.—The *main battery* consists of fourteen 6-inch rapid-fire rifles, 50 calibers in length. Two of these are mounted, one forward and one aft, on the main deck outside the superstructure, on the center line, at frames 23 and 91, respectively, each having a total arc of train of 290 degrees. Four are mounted on the main deck in armored sponsons within the corners of the superstructure, with an arc of train of 90 degrees forward and 35 degrees abaft the beam for the forward pair and 35 degrees forward and 90 degrees abaft the beam for the after pair. The remaining eight are mounted on the gun deck within the upper casemate, in box battery, four on each broadside, with separating splinter bulkheads, as before mentioned; the forward and after pairs of this arrangement are mounted in sponsons at the corners of the upper casemate directly below the corresponding pairs on the main deck, and have a total arc of train of 120 degrees, 85 forward and 35 abaft the beam for the forward pair, and 35 forward and 85 abaft the beam for the after pair; the total arc of train of the second and third pairs of this arrangement is 112 degrees, 77 forward and 35 abaft the beam for the second pair, and 35 forward and 77 abaft the beam for the third pair. The mean height of the axis of main-battery guns above the 22 feet 6 inches L.W.L. is about 21 feet 9 inches for the main-deck battery and 14 feet 6 inches for the gun-deck battery.

The *secondary battery* consists of eighteen 3-inch, rapid-fire guns 50 calibers in length; twelve 3-pounder semi-automatic guns; four 1-pounder heavy automatic guns; eight 1-pounder heavy rapid-fire guns; two 3-inch field guns; two machine guns, .30-caliber; and eight automatic guns, .30-caliber. Twelve 3-inch guns are mounted on the gun deck, outboard, outside the upper casemate, in frame spaces 13–14, 23–24, 31–32, forward, and 82–83, 91–92, 101–102, aft, in pairs, port and starboard. These twelve guns have 3-inch armor port-plates, as before mentioned. The remaining six 3-inch guns are mounted outboard on the main deck within the superstructure, at frame spaces 52–53, (on) 57, 61–62, in pairs, port

and starboard. These guns have no armor-port plates. The 3-pounder guns are equally divided on each side of the vessel and so disposed in commanding positions as to give the largest arc of fire without interference with each other or with the other guns of the battery. Under the original disposition of the battery two 1-pounder heavy automatic guns were to be mounted in each lower military top, fore and main; two 1-pounder heavy R.F. guns in each upper military top, fore and main; two of the remaining four 1-pounder heavy R.F. guns aft on the gun deck; and two 1-pounder heavy R.F. guns forward on the upper deck. A new disposition is now to be given to the guns formerly assigned to locations in the military tops in order to devote these tops to range-finding purposes. The field, automatic and machine guns have field-carriage mounts and cage-stand or rail-socket positions.

BATTERY ARRANGEMENT AND TRAIN.

MAIN BATTERY.

Fourteen 6-inch, 50-caliber R. F. guns.

Number.	Location.	Train, forward and abaft beam.
Nos. 1 and 2.	Gun deck, frame 44, S. and P.,	85 ° fwd., 35 ° aft.
Nos. 3 and 4.	Gun deck, frame 52½, S. and P.,	77 ° fwd., 35 ° aft.
Nos. 5 and 6.	Gun deck, frame 61½, S. and P.,	35 ° fwd., 77 ° aft.
Nos. 7 and 8.	Gun deck, frame 70, S. and P.,	35 ° fwd., 85 ° aft.
No. 9.	Main deck, frame 23, central,	ahead to 55° aft, P. and S.
Nos. 10 and 11.	Main deck, frame 44, P. and S.,	90 ° fwd., 35 ° aft.
Nos. 12 and 13.	Main deck, frame 70, P. and S.,	35 ° fwd., 90 ° aft.
No. 14.	Main deck, frame 91, central,	astern to 55° fwd., P. and S.

SECONDARY BATTERY.

Eighteen 3-inch, 50-caliber, R. F. guns.

Nos. 1 and 2.	Gun deck, frame 13½, S. and P.,	90 ° fwd., 24 ° aft.
Nos. 3 and 4.	Gun deck, frame 23½, S. and P.,	77 ° fwd., 43 ° aft.
Nos. 5 and 6.	Gun deck, frame 31½, S. and P.,	73 ° fwd, 41 ° aft. 76 ° fwd., 44 ° aft.
Nos. 7 and 8.	Gun deck, frame 82½, S. and P.,	42½° fwd., 75 ° aft. 43½° fwd., 73½° aft.
Nos. 9 and 10.	Gun deck, frame 91½, S. and P.,	43 ° fwd., 77 ° aft. 43½° fwd., 76 ° aft.
Nos. 11 and 12.	Gun deck, frame 101½, S. and P.,	24 ° fwd., 90 ° aft.
Nos. 13 and 14.	Main deck, frame 52½, S. and P.,	60 ° fwd., 60 ° aft.
Nos. 15 and 16.	Main deck, frame 57, S. and P.,	60 ° fwd., 60 ° aft.
Nos. 17 and 18.	Main deck, frame 61½, S. and P.,	60 ° fwd., 60 ° aft.

Twelve 3-pounder, semi-automatic guns.

Nos. 1 and 2. Upper deck, frame 28–29, S. and P.
Nos. 3 and 4. Upper deck, frame 30–31, S. and P.
Nos. 5 and 6. Upper deck, frame 45, S. and P.
Nos. 7 and 8. Upper deck, frame 69, S. and P.
Nos. 9 and 10. Upper deck, frame 81–82, S. and P.
Nos. 11 and 12. Upper deck, frame 86, S. and P.

Twelve 1-pounder, heavy, R. F. guns.

Nos. 1 and 2. Gun deck, frame 110¼, S. and P., after cabin.
Nos. 3 and 4. Upper deck, frame 36, S. and P.
*Nos. 5 and 6. Forward lower military top.
*Nos. 7 and 8. After lower military top.
*Nos. 9 and 10. Forward upper military top.
*Nos. 11 and 12. After upper military top.

Eight .30-caliber Colt automatic guns.

Nos. 1 and 2. Upper deck, frame 78–79, S. and P.
Nos. 3 and 4. Forward bridge, frame 29, S. and P.
Nos. 5 and 6. After bridge, frame 82, S. and P.
Nos. 7 and 8. After bridge, frame 85, S. and P.

Two machine guns (Gatling), .30-caliber.

Nos. 1 and 2. Forward bridge, frame 31, S. and P.

Two 3-inch field pieces.

Nos. 1 and 2. Stowed on upper deck, frame 41–42, S. and P.

Ammunition Hoists.—There are twenty-seven electrically-operated ammunition hoists fitted to furnish an adequate supply of ammunition to the battery. Frictional slip gears are provided between the electric motors and the hoists to prevent damage being done to either in case of sudden stoppage of the hoists. Stops are fitted to prevent ammunition falling back in the hoists, these stops being hinged to turn up out of the way when striking ammunition below. The motors are reversible and operate at about 1,000 R.P.M., all necessary rheostats and switches being provided. Gear for operating the hoists by hand is also fitted. Ammunition for the guns in the fighting tops was to be whipped to the tops from the upper deck by electric whip hoists with specially-fitted swinging davits.

* Guns marked (*) are subject to re-location on deck.

Ammunition is to be delivered at the following rates:

For each 6-inch gun	6 complete rounds every minute.
For each 3-inch gun	15 complete rounds every minute.
For each 3-pdr. gun	30 complete rounds every minute.
For each 1-pdr. R.F. gun	30 complete rounds every minute.
For each 1-pdr. aut. gun	100 complete rounds every minute.

Ammunition hoists, 6-inch, number and location:

No. 1 6-inch gun supplied by hoist No. 11, gun deck, frame 47, S.
No. 2 6-inch gun supplied by hoist No. 10, gun deck, frame 47, P.
No. 3 6-inch gun supplied by hoist No. 13, gun deck, frame 57, S.
No. 4 6-inch gun supplied by hoist No. 12, gun deck, frame 57, P.
No. 5 6-inch gun supplied by hoist No. 15, gun deck, frame 58, S.
No. 6 6-inch gun supplied by hoist No. 14, gun deck, frame 58, P.
No. 7 6-inch gun supplied by hoist No. 19, gun deck, frame 67, S.
No. 8 6-inch gun supplied by hoist No. 18, gun deck, frame 67, P.
No. 9 6-inch gun supplied by hoist No. 4, main deck, frame 27, P.
No. 10 6-inch gun supplied by hoist No. 8, main deck, frame 41, P.
No. 11 6-inch gun supplied by hoist No. 9, main deck, frame 41, S.
No. 12 6-inch gun supplied by hoist No. 16, main deck, frame 66, P.
No. 13 6-inch gun supplied by hoist No. 17, main deck, frame 66, S.
No. 14 6-inch gun supplied by hoist No. 25, main deck, frame 88 CL.

No. 1 3-inch gun
No. 2 3-inch gun — supplied by hoist No. 1, gun deck, frame 23, S.
No. 3 3-inch gun — supplied by hoist No. 2, gun deck, frame 23, P.
No. 4 3-inch gun
No. 5 3-inch gun — supplied by hoist No. 3, gun deck, frame 26, S. (these hoists near center line.)
No. 6 3-inch gun

No. 7 3-inch gun
No. 8 3-inch gun — supplied by hoist No. 24, gun deck, frame 87, C.
No. 9 3-inch gun — supplied by hoist No. 26, gun deck, frame 94, P.
No. 10 3-inch gun
No. 11 3-inch gun — supplied by hoist No. 27, gun deck, frame 94, S. (these hoists on and near center line.)
No. 12 3-inch gun

No. 13 3-inch gun
No. 14 3-inch gun — supplied by hoist No. 5, main deck, frame 29, P.
No. 15 3-inch gun — supplied by hoist No. 22, main deck, frame 84, P.
No. 16 3-inch gun
No. 17 3-inch gun — supplied by hoist No. 23, main deck, frame 84, S. (these hoists near center line.)
No. 18 3-inch gun

All 3-pdr. and 1-pdr. guns forward are supplied by hoists Nos. 6 and 7 on upper deck at frame 34, P. and S., those aft being supplied by hoists Nos. 20 and 21, at frame 84, P. and S.

Ground Tackle.—The following anchor equipment is supplied:

Two bower anchors, Baldt type, weight (two) 23,875 pounds.
One sheet anchor, Navy type, weight, 13,076 pounds.
One stern anchor, Navy type, weight, 3,487 pounds.
One stream anchor, Navy type, weight, 2,438 pounds.
Two kedge anchors, Navy type, weight, 815 and 404 pounds.
There are 360 fathoms of Navy type, 2½-inch cable supplied, two bower chains, 120 fathoms each, and one sheet chain, 120 fathoms; the total weight of these cables being 132,000 pounds.

PROPELLING MACHINERY.

There are two four-cylinder, vertical, inverted, direct-acting, triple-expansion propelling engines placed abreast, each in its own watertight compartment, separated by a center-line bulkhead, through which is a communicating watertight door. The engines are right-and-left and are arranged for inboard-turning propellers when going ahead. The designed power of these main engines is 21,000 I.H.P. at about 133 revolutions per minute, with a steam pressure of 250 pounds at the high-pressure valve chest.

The main-engine *cylinders*, four for each engine, consist of casings of cast iron, which include the valve chests, steam ports and passages, lower heads, brackets, tie-rod and support facings, and are machined for all necessary fittings. All cylinders and valve chests are fitted with *working liners* of hard, close-grained cast iron. The cylinder liners have bearings at both ends and at about mid-length, and are secured to the cylinder casings by countersunk screws through the bottom flanges, the casing being machined to take this lower seating of the liners. There is a composition gland at the top end of the liners fitted for packing between the liner and the casing, and which allows for expansion while making a tight joint at the end. The space between the liner and the casing forms a steam-jacket space for the barrel of the cylinder. These liners are bored to accurate dimensions after being fitted and while in their natural vertical position, and are counter-bored to give the piston-packing rings a short override at the ends of the working bore. The valve-chest liners have bearings at the ends, the liner for each valve chest being in two parts, and are

forced into place and secured by screws tapped half into the liners and half into the casings. The *cylinder covers* are of cast iron, strongly ribbed, fitted with central manholes, manhole plates and lagging covers, the I.P. and L.P. covers being fitted to be steam jacketed. The *valve-chest covers* are of cast iron, strongly ribbed, the upper covers containing the balance piston cylinders and the lower covers being bored and faced to receive the valve-stem crosshead guides and stuffing boxes. The valve chests are arranged to draw the H.P. and I.P. valves from the top for examination and the L.P. valves from the bottom.

The *order of cylinders*, beginning forward, is as follows: Forward low pressure, high pressure, intermediate pressure, after low pressure. The crank shafts are forged with the forward low-pressure crank and the high-pressure crank (which form the forward section of the complete crank shaft) opposite each other, and with the intermediate-pressure crank and the after low-pressure crank (which form the after section of the complete crank shaft) opposite each other, the first-mentioned pair being set at right angles to the last-mentioned pair. The *crank sequence* when the engines are going ahead is then: high-pressure, intermediate-pressure, forward low-pressure, after low-pressure, with a uniform angularity of 90 degrees apart.

To provide for the *framing and bracing* of the main engines the four cylinders of each engine are supported on twelve forged nickel-steel cylindrical columns, having swells and facings for the athwartship and longitudinal braces and tie rods, and flanges at top and bottom for securing to the cylinders and bed plates. Athwartships, between each pair of front and back columns, is fitted an X-brace, with a horizontal tie rod through the center of the X, on which are lateral distance pieces. Longitudinal stiffness is provided by diagonal tie rods on the outside of columns, front and back, one from top of first to foot of third column, bolted at center boss of second, and a similar one for the fourth, fifth and sixth columns. On the inside of front columns at center is secured a horizontal fore-and-aft tie rod, having an

adjusting screw and sleeve at its midlength. On the inside of back columns at center is fitted a horizontal I-beam stiffener, to which the lower ends of the crosshead guides are secured. Additional stiffness is given to the engines by athwartship ties between pads on the cylinder bases of the two engines and by fore-and-aft ties secured to pads in the hull structure forward and abaft the engines. This arrangement gives a very stiff and rigid engine, as was shown by the severe test during the official trials.

The engine *bed plates*, consisting of two longitudinal and six cross girders of I section, are of cast steel, three sections for each engine, bolted together at flanges, and each section carrying three main bearings. The bed plates rest on and are secured to the keelson plates provided in the hull structure. Proper facings and finishings are provided for flanges of engine columns, seatings for bearing brasses and caps, and for turning engine.

All the *main steam valves* are of the single-ported piston type, with single cast-iron packing rings and steel followers. The H.P. valves are of cast iron; the I.P. and L.P. valves are built up with two heads of cast iron with a steel-tube distance piece between. The H.P. cylinder has one valve and the I.P. and each L.P. cylinder two each. There is a steam piston at the top of each valve stem, working in a short cylinder in the valve-chest cover, which is so designed as to relieve a part of the weight of the valve gear. The H.P. and I.P. balance pistons are 7 inches in diameter and the L.P. 9 inches in diameter. Steam is taken on the inside of the H.P. valves and on the outside of the I.P. and L.P. valves.

The *valve gear* is of the Stephenson type with double-bar forged-steel links, the H.P. valve stem of each engine being connected directly to its link block while the pairs of I.P. and L.P. valve stems for their respective cylinders are connected by a crosshead.

The maneuvering of the engines is controlled by a steam *reversing engine* for each, of the floating-lever type, the piston rod of which is connected to the reverse shaft carried on the

front engine columns, and also carries a control piston in an oil cylinder giving steadiness of motion and allowing for hand operation by means of an oil pump. Independent *cut-off blocks* are fitted, one for each link, in the arm of the reverse shaft, giving a range of independently adjustable cut-off from .5 to about .73 of the stroke.

All main-engine cylinders are *steam jacketed* around the working liners, and the I.P. and L.P. cylinders are also jacketed at the ends. Live steam from the main steam pipe on the boiler side of the engine stop valves is led through the H.P. jackets and thence through the I.P. and L.P. jackets, with suitable reducing and safety valves.

Each engine is fitted with a balanced *throttle valve* having a clear opening equal to 13 inches in diameter, the balance being accomplished by a piston working in a cylinder cast in the valve casing.

The *pistons* are conical in shape, the H.P. being of cast iron and the I.P. and L.P. of cast steel. The H.P. and I.P. pistons are each fitted with one hard cast-iron packing ring $2\frac{1}{2}$ inches wide and 1 inch thick, arranged for limited expansion and set out by seven U-shaped springs in the joint. There are also eighteen double-ended, flat-looped springs for setting out each H.P. packing ring, thirty-three for each I.P. ring and thirty-nine for each double L.P. ring. The L.P. pistons are each fitted with two rings $1\frac{1}{4}$ inches by 1 inch, each separated by diagonal cuts into eight sections. Composition tongue pieces, set in flush, cover these cuts between the sections and the sections of one ring break joint with those of the other, dowels preventing independent rotation of one ring on the other.

The *follower rings* of all pistons are of cast steel, secured to the piston body on square-shouldered studs by nuts and split pins. All pistons have $14\frac{1}{2}$-inch bosses with outside threads for lifting and handling, and are taper-bored to receive the piston-rod end, setting up against a filleted collar on the rod, and secured from turning by a dowel in the rod. The piston rod is threaded beyond the face of the piston boss and a cylindrical nut, slotted for a spanner, sets up against the boss face

and is locked by a ring with tongues corresponding to the slots in the nut which slips over the nut and is bolted to the boss face.

The *piston rods* are hollow, of nickel-steel, oil-tempered and annealed, accurately ground and polished to 8 inches outside diameter, the axial hole being 4 inches in diameter for the H.P. and I.P. rods and 5 inches for the L.P. rods, except at the crosshead ends where it tapers to $1\frac{1}{2}$ inches for the H.P. and I.P. and $1\frac{3}{4}$ inches for the L.P. rods respectively.

The *crossheads* are of forged steel, taper-bored to receive the piston rod, and machined for the crosshead slipper of cast steel which is bolted to the flanged lug of each. The piston rod is threaded at the end beyond the face of the crosshead, and a cylindrical nut, slotted for a spanner, sets up against the bottom face of the crosshead. The crosshead pins extend on each side of the crosshead, being forged in one piece with it, are taper-bored from ends to center of crosshead, and have a $\frac{3}{8}$-inch flat on the sides.

The *crosshead guides* are of cast iron, hollow for the circulation of water, and are secured by their bottom ends to the strongbacks on the back engine columns and at their tops to lugs on the cylinder casings. They are machined with side flanges to which the L-shaped backing guides are bolted.

The *connecting rods* are of nickel-steel, oil-tempered and annealed, having a 4-inch axial hole for the H.P. and I.P. rods and a 5-inch hole for the L.P. rods. The upper end is forked to span the crosshead and the ends of the fork are machined to take the crosshead brasses which are bolted to them with two bolts each. The lower end is machined to take the crank-pin brasses, which are bolted to it with two bolts, the bottom brass being backed by a machined, forged, flat-steel cap.

The *eccentrics* are made in two parts, the thin section of forged steel and the thick section of cast iron. These are accurately fitted together and secured by two 2-inch bolts. After being fitted to their seating they are turned on the outside to a uniform eccentricity of 5 inches. The backing eccen-

tric of each pair is keyed to the shaft and the go-ahead eccentric is bolted to the backing eccentric by two through bolts in elongated holes, the holes being filled in solid after setting the eccentrics to prevent displacement.

The *eccentric straps* are of composition, lined with white metal, accurately fitted to the eccentrics.

The *eccentric rods* are of nickel-steel, having T heads at the lower ends for securing to the eccentric straps and forked-top ends to span the main links at the link pins, adjustable brasses being fitted for these pins.

The *main links* are nickel-steel curved bars, having the pins for suspension rods and eccentric rods forged on. Each pair of bars is securely bolted together at the ends, and the ahead half of the link is divided into ten equal divisions, from 0, the mid-position, to 10, full gear.

The *main link blocks* are of nickel-steel forgings, fitted with jaws at the ends to span the bars of the links, these jaws being provided with adjustable composition gibs conforming to the curve of the links.

All *shafting* is hollow and is of forged nickel-steel. The crank shafts have two sections, and the thrust, line, stern-tube and propeller shafts have one section each for each main engine. A worm wheel for turning the main engine is fitted over the coupling flanges between the two sections of crank shaft, one for each engine. There are six *main bearings* for each complete crank shaft, two *steady bearings* for each thrust shaft, one *spring bearing* for each line shaft, two *stern-tube bearings* for each stern-tube shaft, and one *strut bearing* for each propeller shaft. Each shaft is made watertight at the forward end of its stern tube by a stuffing box and gland having a packing space 7 inches deep by 1 inch wide, arranged to prevent rotation of the packing with the shaft, and fitted with a water-circulation pipe from behind the packing space to the engine-room bilge. A stuffing box and gland of less depth is fitted around the shaft at the bulkhead between the engine room and shaft alley.

The *thrust bearing* is of the horseshoe type, the pedestal being of cast iron bolted down, through oblong holes, on a cast-iron sole plate which is fitted with raised lugs at the ends which are machined for fore-and-aft adjusting wedges fitted in the space between the ends of the pedestal and these lugs. The sole plate is in turn riveted solidly to the built-up foundation from the hull structure. The sides and ends of the pedestal form a trough for oil in which the lower part of the thrust collars turn, the trough being fitted with a closed water-circulating coil for cooling the oil. The boxes for the steady bearings which carry the weight of the thrust shafts are cast in one with the pedestal as are also the lugs supporting the side adjusting rods, and a stuffing box is worked around the shaft at each end of the pedestal to prevent the escape of oil. There are twelve horseshoes of cast steel, made hollow for the circulation of water through each separately, and lined with white metal on both sides, except the forward one, which is for ahead thrust only and bears only on the forward side of the forward collar. Provision is made for independent fore-and-aft adjustment of the individual horseshoes by 4-inch, steel, screwed side rods and wrought-iron clamp nuts.

The *inboard coupling* of each stern-tube shaft consists of a forged-steel collar secured to a raised seating on the end of the shaft by four steel keys; behind this solid collar is another steel collar in halves and fitting snugly around the shaft behind the shoulder of the collar seating, both collars being through bolted to the after flange of the line shaft by eight 3¼-inch fitted bolts.

The *outboard coupling* between stern tube and propeller shafts consists of a forged-steel solid sleeve, taper bored from ends to center, secured from turning on the tapered shaft ends by four fitted keys for each shaft (two for each shaft end). The shaft ends are forced into the taper bores with the keys in their seats, and tapered drawing cotters are driven through slots in sleeve and shaft ends to prevent withdrawal, the grooves around cotter slots being filled with solder and screw stop pins being let in to secure the cotters.

ENGINE DATA.

Cylinders, number for each engine	4
H.P., diameter, inches	36
I.P., diameter, inches	59.5
L.P. (2), diameter, inches	69
Stroke of all pistons, inches	45
Valves, H.P. (1 each H.P. cylinder), diameter, inches	22
I.P. (2 each I.P. cylinder), diameter, inches	25
L.P. (2 each L.P. cylinder), diameter, inches	$27\frac{1}{2}$
Valve stems, diameter, inches	$3\frac{1}{4}$
through valve, inches	$2\frac{1}{4}$
above valve, inches	$1\frac{3}{4}$
at balance piston, inches	$1\frac{1}{2}$
Main steam pipe, diameter at throttle, inches	13
area of cross section, square inches	132.73
Volume swept by H.P. piston, mean, per stroke, cubic feet	26.073
I.P. piston, mean, per stroke, cubic feet	72.409
L.P. pistons (2), mean, per stroke, cubic feet	194.754
Net area of I.P. to H.P. piston, ratio	2.7316 to 1
L.P. pistons (2) to H.P. piston, ratio	7.3373 to 1
Piston rods, diameter, inches	8
of axial hole, H.P. and I.P., inches	4
L.P., inches	5
length, feet and inches	9- $2\frac{1}{2}$
Cylinder walls, thickness, H.P., I.P., L.P., inches,	$2\frac{1}{8}$, $1\frac{3}{4}$, $1\frac{1}{2}$
liners, thickness, H.P., I.P., L.P., inches	$1\frac{1}{8}$, $1\frac{7}{16}$, $1\frac{1}{8}$
Valve-chest liners, thickness, H.P. and I.P., inches	$1\frac{1}{8}$
L.P., inches	$1\frac{5}{16}$
Jacket space, inch	$\frac{7}{8}$
Cylinder relief valves (2 each cylinder), diameter, inches	4
Valve-chest relief valves (1 each I.P. and L.P. valve), diameter, inches	4
Connecting rod, length center to center, inches	90
diameter, upper end, inches	$8\frac{1}{2}$
lower end, inches	$9\frac{3}{8}$
axial hole, H.P. and I.P., inches	4
L.P., inches	5
Connecting-rod crosshead bolts (4), diameter, inches	$3\frac{1}{4}$
axial hole, diameter, inches	$1\frac{1}{2}$
crank-pin bolts (2), diameter, inches	$4\frac{1}{2}$
axial hole, diameter, inches	$1\frac{7}{8}$
Crosshead slipper, surface, ahead, square inches	630
backing, square inches	472.5
pin, diameter, inches	$10\frac{1}{2}$
of axial hole, tapered ends to center, ins.	6 to 2
length over all, inches	$35\frac{3}{8}$
each side crosshead, inches	$10\frac{1}{2}$
Crank shaft, number of sections each engine	2

Crank shaft, diameter, inches	18
axial hole, inches	11¼
coupling discs, inches	32
thickness coupling discs, inches	3⅜
coupling bolts (8 each flange), tapered, diameter, ins.	3 9/16
journals, diameter, inches	18
length, inches	(4) 21¼, (1) 31¼, (1) 26½
length forward section, F.L.P. and H.P., ft. and ins.	15–11½
after section, I.P. and A.L.P., feet and ins.	17– 8½
distance between journal centers, ft. and ins., 5–6, 5–6, 5–11, 6–1⅝	
pin, diameter, inches	18¼
length, inches	22
projected area, square inches	407
axial hole, diameter, inches	12
webs, width, inches	20⅝
thickness, inches	11
Thrust shaft, diameter, inches	17½
axial hole, inches	11½
length, feet	15
steady bearings, inches	18
collars, number on each shaft	12
diameter, inches	27
thickness, inches	2
space between, inches	4
horse shoes, number, each engine	12
total ahead surface, each engine, square inches	2,568
backing surface, each engine, sq. inches	2,354
steady bearings (2 each), length, inches	18
diameter, inches	17½
Line shaft, diameter, inches	17
axial hole, inches	11½
coupling disc, forward, inches	32
after, inches	33½
bolts (8), 1 flange, inches	3½
length, feet and inches	21– 3
Stern-tube shaft, diameter, inches	18
axial hole, inches	11
length, feet	36
Propeller shaft, diameter in strut bearing, inches	18
axial hole, inches	11
length, feet and inches	34– 6
Inboard collar coupling, diameter, inches	33½
length, inches	17½
of collar, inches	14
keys (4 each shaft), inches	14 × 2½ × 1½
Outboard coupling, diameter, inches	23½
taper to center, 1 inch per foot, inches	18 to 15½
keys (4 each shaft), inches	32½ × 2½ × 1½

Outboard coupling, through-gibs (2 each shaft), inches, $23\frac{1}{2} \times 2\frac{1}{2} \times 7\frac{1}{2}$ to $6\frac{1}{2}$	
length over all, feet and inches	5-10½
Stern-bracket bearing, diameter, inches	20½
length, feet	6
Stern-tube bearing, after, diameter, inches	$20\frac{1}{16}$
length, feet	6
forward, diameter, inches	20
length, feet	6
Reversing engine, diameter of steam cylinder, inches	16
oil cylinder, inches	8
stroke, inches	22
Turning engine, diameter of steam cylinders (2), inches	7
stroke, inches	5

Main Condensers.—There is one main condenser for each engine, located abreast of engine, outboard. The shell is of steel, butted with double butt straps and stiffened with T bars circumferentially, and fitted with riveted angle-iron rings at the ends to form flanges for securing the water chests. The water chests are of composition and are partially lined with rolled-zinc plates. At the inlet and discharge end a division plate is placed horizontally across the water chest to direct the flow of water through the tubes, and in this plate is fitted a valve, worked by a lever on the outside of the condenser, to allow the circulating water to pass directly overboard when this valve is opened. Steel baffle plates, fitted directly under the inlets, deflect the entering steam and distribute it over all the tubes. The tubes are of seamless-drawn Admiralty metal, not tinned, packed with brass screw glands and cotton tape, and supported by two plates, spaced 4 feet 7 inches from the tube plates, through which the tubes are passed, the tubes resting in metal thimbles of the same composition as the tubes. Two main inlet nozzles are riveted to the top of the condenser, one at each end, on the forward one being arranged seatings for the various accessory inlets. At the bottom of the condenser is a 12-inch suction opening to the main air pump and a 12-inch manhole, in the cover of which is a boss for a boiling-out connection. In the bottom of each tube plate is a hand hole to facilitate examination, and the water-chest covers are fitted with circular manholes to allow examination of tube

CYLINDER CLEARANCE.

Engine.	Cylinder.		Volume of cylinder, cubic inches.	Clearance, in gallons of water.	Clearance, in cubic inches.	Per cent. of cylinder volume.	Linear clearance.
Starboard, . .	H.P.	Top.	45,805	55	12,705	27.7	$1\frac{5}{32}$
		Bottom.	43,543	59.25	13,687	31.4	$1\frac{3}{32}$
	I.P.	Top.	125,123	123	28,413	22.7	$\frac{9}{16}$
		Bottom.	122,861	129.5	29,914	24.3	$\frac{9}{16}$
	F.L.P.	Top.	168,268	135.5	31,300	18.6	$\frac{5}{8}$
		Bottom.	166,006	145.5	33,610	20.2	$1\frac{9}{32}$
	A.L.P.	Top.	168,268	134	30,954	18.4	$1\frac{9}{32}$
		Bottom.	166,006	149	34,419	20.7	$\frac{3}{4}$
Port, ...	H.P.	Top.	45,805	56	12,936	28.2	$1\frac{7}{32}$
		Bottom.	43,543	61	14,091	32.3	$1\frac{7}{32}$
	I.P.	Top.	125,123	120	27,720	22.1	$\frac{7}{16}$
		Bottom.	122,861	131.5	30,376	24.7	$\frac{3}{4}$
	F.L.P.	Top.	168,268	135	31,185	18.5	$\frac{9}{16}$
		Bottom.	166,006	153.5	35,458	21.3	$\frac{3}{4}$
	A.L.P.	Top.	168,268	130.5	30,145	17.9	$1\frac{3}{32}$
		Bottom.	166,006	155	35,805	21.6	$1\frac{5}{32}$

ends and packings. Water and vacuum spaces are provided with all necessary drains.

Diameter of shell, inside, feet	7
Thickness of shell, inch	$\frac{5}{8}$
Length over heads, feet and inches	17- 1
Tubes, diameter, outside, inch	$\frac{5}{8}$
length between tube sheets, feet and inches	14- 6
as ordered, over all, feet and inches	14- $8\frac{1}{2}$
thickness, No. 18 B.W.G., inch	.048
spacing between centers, inch	$\frac{11}{16}$
number in each condenser	5,684
Cooling surface, both condensers, square feet	26,966

Main Air Pumps.—For each main engine there is a Blake, twin, vertical, beam, single-acting air pump, having two steam and two water cylinders. On the top of a lower suction base are the suction valves, eighteen in number for each pump barrel, and in the pump pistons and in the discharge chamber are the bucket and discharge valves respectively, eighteen each for each pump barrel. These valves are all uniform in size and construction, consisting of three bronze discs of graduated sizes, of which the top disc is plain except for the stud hole, while the other two discs have small holes within the rim of the overlying disc to allow the escape of water between the separate discs. The discs are held against their seats by a guard and spiral springs secured to the central stud. Examination of the suction and bucket valves is provided for by removable manhole covers which form a part of the pump barrel, the bucket being placed at the top and bottom positions respectively. Examination of the delivery valves is made through hand holes in the delivery chests, and these delivery valves have also a removable seating in contrast with the suction and bucket valves where the seating is a part of the valve plates.

Diameter of steam cylinders (2), inches	14
pump cylinder (2), inches	35
Stroke, inches	18
Diameter of piston rods, nickel-steel, inches	$2\frac{1}{2}$
pump rods, Tobin bronze, inches	$3\frac{1}{2}$
valves in bucket, head and foot, inches	$5\frac{1}{2}$
Number of valves (18 each bucket, head and foot), each barrel,	54

Number of valves one complete pump, both barrels	108
Area through 18 valve openings, square inches	234
Lift of valves, possible under guards, inch	$\frac{11}{32}$
Discharge (area × lift), square inches	153.55

Main Circulating Pumps.—For each engine there is one centrifugal, double-inlet, circulating pump driven by a two-cylinder, compound, vertical, inverted engine. The specified capacity of each of these pumps is 12,000 gallons per minute when discharging from the bilges when running at not more than 266 revolutions per minute. Each pump draws from the sea, the main drain and from the engine-room bilge, and discharges to the main condenser, through its tubes and overboard when performing the regular duties, or directly overboard through the condenser by-pass, if desired, when in use on the bilges. The suction valves are so locked with inter-connecting gear that no two of them may be open at the same time.

Diameter of steam cylinders, inches	H.P., 11; L.P., 22
Stroke, inches	10
Diameter of pump runner, inches	45
Width of pump runner at periphery, inches	4
hub, inches	$10\frac{9}{16}$
Diameter of inlet nozzles (2), inches	$15\frac{1}{4}$
outlet nozzle, inches	21

Feed and Filter Tanks.—Abaft the main condenser in each engine room is located a combined feed and filter tank of about 5,330 gallons capacity. The after end of the tank, higher than the forward part, is arranged as a filter and grease extractor, and is in communication with the lower part of the tank over the top of a weir at the back. Filtering material is packed loosely in the three compartments of the filter section of the tank, and all discharges from main and from auxiliary air pumps are so led that the water must pass three times downward through the filter sections in succession before reaching the feed tank proper. Vapor pipes connect to the atmosphere from the top of the tanks, and all necessary gauge glasses, drains, overflow outlets and suction connections are provided, as well as bolted covers and manhole plates, for the proper access to all parts. The suction outlets to the hotwell pumps

and to the cross connection from which the feed pumps take their suctions are protected from inrush of air when water is low by a valve controlled by a float.

Main Feed Pumps.—There are four main feed pumps, one at the starboard side of each of the four athwartship firerooms, of the Blake, single-cylinder, double-acting, plunger type. The suction for these pumps is from the feed tanks direct (from the cross-connecting pipe between the feed tanks) and from the main into which the hotwell pump discharges through the feed heaters, no other suctions being provided. The discharge from these pumps is to the main feed line only.

Auxiliary Feed Pumps.—The auxiliary feed pumps are duplicates of the main feed pumps and are placed on the opposite side of the firerooms, four in number. The suction for the auxiliary feed pumps is from the feed tanks (through cross connection as for main feed pumps), from the auxiliary drain, from the sea, from the floor of its own compartment and from the bottom blow pipes of all boilers in its own compartment. (It is possible, also, to divert the feed water after passing through the feed heaters to enable the auxiliary feed pumps to handle it.) The discharges of these pumps are: to the feed main for all boilers, to the fire main, overboard through sea valve in their own compartments and through special fittings for fireroom purposes.

Main and auxiliary feed pumps, four each, center-packed plungers:

Diameter of steam cylinder, inches	14
water cylinder, inches	10
piston rods, inches	2¼
Stroke, inches	15
Valves, suction and delivery, 10 each per pump, diameter, inches	3¼
area of opening (1), square inches	5¼

Feed-Water Heaters.—In each engine room is a feed-water heater of 950 square feet of heating surface, so arranged that the discharge of the hotwell pumps may be passed through the heater or be by-passed, the auxiliary exhaust steam being used to heat the feed water on its way through the heaters. The design provides that the heating steam is on the outside

of the tubes, being distributed by baffles, which also serve as supporting plates for the tubes, while the water passes into the lower half of the tubes and leaves through the upper half by way of suitable nozzles on the inlet and outlet head. The pressure of the auxiliary exhaust steam for the heaters is regulated by relief valves on the connections from the exhaust line.

Diameter of shell, inside, feet and inches	2–11¼
Thickness of shell, inch	¼
Length over heads, feet and inches	8– 9
Tubes, diameter, outside, inch	⅝
length between tube sheets, feet and inches	6–10
total, as ordered, feet and inches	6–11¾
thickness, No. 16 B.W.G., inch	.065
spacing between centers, inch	1⅛
number in each heater	852
Heating surface, both heaters (950 each), square feet	1,900

Hotwell Pumps.—There is a hotwell pump in each engine room with suction connections for drawing water from the main feed tanks, from the reserve feed tanks in inner bottoms, from either main air-pump suction pipe, and from the fresh-water connections at the ship's side for taking in fresh water from lighters alongside. There are discharge connections through the feed heaters into the feed suction mains from which the main feed pumps draw, into the feed tanks, and into the reserve feed tanks.

Distiller Circulating Pump.—In the starboard engine room is a distiller circulating pump, a duplicate of the hotwell pumps, having a suction to the sea only, and discharging either through the distillers or by-passing them, into the fire main, the flushing system, or overboard. Relief valves are fitted on the discharge pipes to the distillers to guard against excessive pressure on the distiller coils. Hotwell (2) and distiller-circulating (1) pumps, Blake light-service piston type.

Diameter of steam cylinder, inches	12
water cylinder, inches	14
piston rods, inches	2½
Stroke, inches	12
Valves, suction and delivery (18 each per pump), diameter, inches	3½
area of opening (1), square inches	5

Fire and Bilge Pumps.—There is one fire and bilge pump in each engine room, Blake, vertical, simplex piston type, each having suctions for drawing water from the sea, the auxiliary drain, the drainage manifolds, the engine-room bilge and the crank pits. They discharge to the fire main and overboard.

Diameter of steam cylinders, inches	12
water cylinders, inches	10
piston rods, inches	2¼
Stroke, inches	12
Valves, suction and delivery (18 each per pump), diameter, inches	2¾
area of opening (1), square inches	2.57

Shaft Bilge Pumps.—On each main-engine shaft there is a bilge pump, worked by a crank pin on the end of the main shaft. These pumps have a capacity of about 390 cubic inches per stroke, and draw water from the engine-room bilges and from the crank pits, discharging overboard.

Auxiliary Condensers.—There is one auxiliary condenser in each engine room and one in the lower dynamo room, starboard side, of the type known as the Wheeler condenser, having a combined air and circulating pump attached to the condenser and located on the foundations under it.

	Engine rooms (2).	Dynamo room (1).
Diameter of shell, inside, inches	28	35¼
Thickness of shell, inch	⅝	⅝
Length over heads, feet and inches	7-10½	9- 1½
Tubes, diameter, outside, inch	⅝	⅝
length between tube sheets, feet and inches	6- 8	7- 4
total, as ordered, feet and inches	6-10½	7- 6½
thickness, No. 18 B.W.G., inch	.049	.049
spacing between centers, inch	1⅛	1⅛
number in each condenser	552	1,001
Cooling surface, one condenser, square feet	601.4	1,200.9
Pump cylinders, steam, diameter, inches	8	12
circulating, diameter, inches	10	14
air, diameter, inches	10	14
piston rods (2), diameter, inches	1½	1⅝
stroke, inches	12	12
Valves, air, suction and delivery, number each per pump	4	4
diameter, inches	5⅝	7
area of opening (1), square inches	13.6	21.5
circulating, suction and delivery, number each per pump	4	6
diameter, inches	5⅝	5⅝
area of opening (1), square inches	13.6	13.6

Screw Propellers.—The propellers are of manganese-bronze, and are arranged to turn inboard when going ahead, the starboard propeller being left-hand and the port one right-hand. They are designed as true screws and are accurately balanced, with pitch adjustable from 20 feet 9 inches to 18 feet 9 inches, the designed pitch being 19 feet 9 inches. Each boss is accurately bored to fit the taper on the end of the shaft, is fitted with two feather keys, and is held by a nut screwed on the end of the shaft against the boss and locked in place. A cap of composition is fitted watertight over the end of the shaft and nut to fair the end of the boss and protect the shaft. The bolt recesses in blade flanges for securing the blades to the seatings in the boss are covered by watertight plates, set in flush with the flange surfaces. Blades, bosses, caps and plates are smoothed, finished and burnished.

Number of blades, each propeller	3
Diameter of propeller, feet and inches	16- 3
hub, feet and inches	4- 6
Length of hub, inches	42
Pitch of propellers, official trial, feet and inches	20- 6
Helicoidal area, each, square feet	80
Disc area, each, square feet	207.395
Projected area, each, square feet	64.5
Center of hub above lowest point of keel, feet	9.2
Top of propeller immersed (22 feet 6 inches draught), inches	62½

After the official trials of the *Charleston* the Bureau of Steam Engineering directed the contractors to make the spare propeller blades for this vessel to conform to the following dimensions: Diameter of propellers, 17 feet 6 inches; helicoidal area, 95 square feet; pitch, 19 feet 6 inches, mean. The form of the new blades is to be similar to that of the originals, and the same range of adjustability of pitch is required; that is, one foot above and below mean pitch.

Boilers.—There are sixteen Babcock & Wilcox boilers, arranged in four groups of four each, in four watertight compartments across the center line between the coal-bunker bulkheads. The two forward boilers of each group are located to starboard and port of the center line, respectively, and face

the corresponding two after boilers of the same group, the athwartship fireroom for the group being between the forward and after pairs. There is a separate smoke pipe for each of the four groups of four boilers each, the vertical axis of the pipes being over the centers of the fireroom spaces for the four groups. The general type and arrangement of these boilers corresponds to those of the *West Virginia* and *Maryland*, of which a detailed description will be found in the illustrated article by Lieutenant R. K. Crank, U. S. Navy, on page 1223 *et seq.*, Vol. XVI, November issue, 1904, of the JOURNAL. The feature of the division of the furnaces of the boilers of those vessels by a water leg of four square-sectioned middle boxes and a 4-inch tube is not found in the boilers of the *Charleston*, where the number of sections making up the width of the boiler is smaller and the width of the furnace is consequently less.

Number of boilers	16
furnaces	16
Steam drum, inside diameter, inches	42
thickness, inch	$\frac{21}{32}$
Floor space, one boiler, length, feet and inches	10- 1
width, feet and inches	13- $8\frac{1}{2}$
Grate, length, feet and inches	7- 0
width, feet and inches	12- 6
surface, one boiler, square feet	87.5
all boilers, square feet	1,400
Heating surface, one boiler, square feet	4,000
all boilers, square feet	64,000
Heating surface ÷ grate surface, ratio	45.71
Tubes, one boiler, 2-inch, No. 8 B.W.G., 9 feet 2 inches long	672
4-inch, No. 6 B.W.G., 9 feet 2 inches long	23
8 feet 11 inches long	2
7 feet 3 inches long	23
Elements, one boiler (21 intermediate, 2 sides)	23
Weight of water, one boiler, at steaming level, pounds	13,785
complete, pounds	80,796
Working pressure, on trials, pounds, per square inch	300
Air pressure on trials, inches of water column	2
Safety valves (1 triple-type each boiler), total	48
diameter of each valve, inches	3
set to blow at, pounds	300
Sentinel valves (1 each boiler), diameter, inch	$\frac{1}{2}$

Sentinel valves set to blow at, pounds	295
Smoke pipes (4), diameter, outside, feet and inches	8– $5\frac{9}{16}$
inside, feet	8
area, each, square feet	50.2655
total, square feet	201.062
height above grates, feet and inches	92–$10\frac{1}{4}$
Escape pipes (4), diameter, inches, after pipe	10
three others	9

Forced-Draft Blowers.—There are eight forced-draft blowers of the Sturtevant type located in recesses in the uptakes and above the protective deck. The arrangement and location of these blowers is such that two blowers discharge into each of the four firerooms, taking their suction from the air spaces around the base of the smoke pipe of the boilers they supply, suitable airtight shutters being fitted over boiler-hatch openings to make the firerooms tight. All blowers are readily accessible from the berth-deck compartments and are well protected from the dust and heat of the firerooms, so that they may be well cared for under forced-draft conditions and may be expected to be far more efficient and durable than in the firerooms. By opening the access doors to the blower engines while in operation the blowers become very serviceable as additional ventilation to the lower-deck spaces by the air drawn into the casings from these spaces.

Forced-draft blowers (8), two-cylinder, double, vertical, enclosed :	
Steam cylinders (2), diameter, inches	7
Stroke, inches	5
Diameter of piston rods, inches	$1\frac{5}{16}$
fan, inches	66
Area of induction nozzle, square inches	1,410.3
eduction nozzle, square inches	1,755

General Workshop.—The following tools are provided and installed in the general workshop on the port side of the berth deck abreast of the engine-hatch enclosure:

One lathe, screw-cutting, back-geared, gap, swinging 30 inches over the ways and taking 10 feet between centers;

One lathe, screw-cutting, back-geared, 14-inch, taking 4 feet between centers;

One column-shaping machine, 16-inch stroke, 24-inch traverse;

One drilling machine, back-geared, to drill up to 1½ inches, 14 inches from edge of work;

One sensitive drill, 16-inch;

One milling machine, universal, No. 1;

One combined punch and shears, hand, 6-inch blades, with capacity for shearing ¾-inch round iron, ⅝-inch steel plate, and punching ⅝-inch holes in ⅝-inch mild-steel plates;

One emery grinder, two wheels, 12-inch diameter and 2-inch face;

One grindstone, 30-inch;

Four bench vises are mounted on two vise benches convenient for hand work, two other vices being mounted on small benches in the engine rooms.

The above power tools are driven by a system of overhead line and counter shafting belted from a 10-H.P. electric motor in the forward outboard corner of the workshop.

Ash Hoists.—Four fireroom ventilators, one in each athwartship fireroom, alternately starboard and port from forward, are fitted with sets of angle-iron guide strips, which guide the ash buckets in their ascent from the fireroom to the main-deck level. These guides are so fitted that they do not interfere with the placing of the armor bars in the protective-deck openings when the ash hoists are not in use. The hoisting is done by four double-cylinder, vertical, reversible steam engines of the Williamson Brothers' servo-motor type, located within the fireroom hatch enclosures at the main-deck level and worked by means of hand wheels and geared shafting from the hoisting station. The engine cylinders (two) are 4½ inches in diameter by 4½ inches stroke; the cranks are set at right angles, and safety gears are provided to prevent overwinding and to stop the buckets when they reach the fireroom floor. From the four hoisting stations the ash buckets are carried by trolley tracks to their corresponding ash chute direct, these chutes being located at frame spaces 48–49 and 58–59, starboard and port, or, by cross-connecting trolley tracks and switches, either of the forward two stations may dump in either of the forward chutes, and either of the after two stations

may dump in either of the after chutes. For dumping into a lighter alongside, the buckets are carried on trucks through the forward doors of the superstructure and dumped in portable chutes secured to the water-ways.

Distilling Apparatus.—There are four evaporators, of the usual Navy type, of a combined capacity of 21,000 gallons per day, and three distillers of the horizontal-tube type, in which the steam passes through the tubes and the cooling water around them, having a combined capacity of 21,000 gallons of potable water per day. The distillers are located in the fireroom-hatch enclosure, forward of the after smoke pipe, at the level of the gun deck, the evaporator room being on the berth deck below them. Aside from the distiller circulating pump before mentioned, there is a connection from a riser to the fire main, from which the circulating water may be supplied to the distillers, using one of the pumps which discharge to the fire main.

Evaporators (4)	3 port side athwartships, 1 starboard side fore-and-aft.
Shells, steel, diameter, inside, feet	5
*thickness, inch	(2) $\frac{5}{8}$, (2) $\frac{1}{2}$
†length over heads and braces, feet and inches	5– 8$\frac{1}{4}$
†total over all, including heads and stiffeners, feet and inches	6– 8$\frac{1}{2}$
Heads, steel, thickness, inch	$\frac{1}{2}$
Tube sheets, composition, thickness, inch	$\frac{3}{4}$
Tubes, brass, No. 12 B.W.G., one evaporator, number	96
outside diameter, inches	2
†length between tube sheets, feet and inches	5– 3$\frac{1}{2}$
†total, as ordered, feet and inches	5– 5$\frac{1}{2}$
†Tube-heating surface, one evaporator, square feet	266
†Capacity, one evaporator, per day, gallons	5,250
Dry-pipe, brass, diameter, inside, inches	5
slots ($\frac{1}{2}$ inch × 130 degrees arc, transverse), number	20
area ($\frac{1}{2}$ × 5.672 × 20), square inches	14.18
Steam connection to coil, diameter, inches	2$\frac{1}{2}$
Drain connection to trap, diameter, inches	1$\frac{1}{4}$
Distillers (3)	horizontal.
Shells, cast iron, diameter, inside, inches	19$\frac{1}{2}$

*Original design for high- and low-pressure evaporators in double effect not changed though single-effect system is now used.

†After port evaporator shortened 4 inches, shell and tubes, to allow withdrawal of coils in space allotted.

Shells, thickness, inch	$\frac{7}{16}$
length over all, feet and inches	5- $8\frac{1}{4}$
Tubes, brass, No. 16 B.W.G., one distiller, number	192
outside diameter, inch	$\frac{5}{8}$
length between tube sheets, feet and inches	4- $5\frac{1}{4}$
total, as ordered, feet and inches	4- 7
Tube cooling-surface, one distiller, square feet	140
Capacity, one distiller, per day, potable water, gallons	7,000
Circulating water, inlet and outlet, diameter, inches	9
Steam inlet, diameter, inches	9
Fresh-water outlet, diameter, inches	$3\frac{1}{2}$

Refrigerating Plant.—One horizontal, two-ton, Allen, dense-air, ice machine is installed in its compartment on the starboard side of the berth deck between frames 73 and $75\frac{1}{3}$, from which refrigerating piping and the necessary returns are taken to the refrigerating rooms, the after scuttle butt and the ice-making tank. The refrigerating rooms are four in number—captain's, junior and warrant officers', wardroom officers', and crew's—with a common air lock from which all open, and are located between frames 68 and 73 on the starboard side of the berth deck. The after scuttle butt is located just forward of frame 73 on the gun deck, starboard side, over the after end of the refrigerating rooms, the forward scuttle butt not being piped with cooling coil. The ice-making tank is located in the after inboard corner of the ice-machine room. The general location of the plant is unfortunate so far as efficiency is concerned owing to the high natural temperature of that portion of the ship and the close proximity of the engines and boilers. As the ice machine is in one unit this will also operate to lower the continued efficiency of the plant through the necessity of long-continued running of the machine with but little chance for overhaul. Another obstacle to equal efficiency of all the rooms lies in the fact that the piping is in combination for the captain's and the wardroom officers' spaces, and for the junior and warrant officers' and the crew's spaces, so that the first mentioned of each of the two combinations must receive the full cooling effect before the second mentioned, and the circulation must be kept up on the whole combination even after one of the two spaces is cooled to the desired

degree. The ice-making tank may be used (a) alone, (b) with rooms, (c) with rooms and scuttle butt, (d) with scuttle butt, or (e) it may be cut out when not required. From the foregoing, with proper substitution of names, may be seen the combinations of uses possible with the plant.

Cylinder, steam, diameter, inches	9½
compressor, diameter, inches	7
expander, diameter, inches	5½
circulating pump, diameter, inches	3½
primer pump, diameter, inches	2
Stroke of crank, inches	13
Piston rods (3), diameter, inches	1¾
length, over all, feet and inches	2- 8½
Cooling coil, face to face of elbows, length, feet and inches	4- 2$\frac{13}{16}$
center to center of ends of coil, length, feet and inches	3- 9$\frac{5}{16}$
complete spiral, turns, number	18½
diameter, mean, inches	12½
tubing, diameter, outside, inches	2
inside, inches	1¾

Anchor Windlass.—The anchor windlass is of the Hyde type, designed for handling four chains, and is located in a pocket between the main and gun decks. Its main worm wheel on the wildcat shaft is driven by a vertical shaft with a worm pinion at its upper end and a bevel gear at its lower end, the latter bevel gear meshing with a similar gear on the after end of the engine shaft. Suitable thrust bearings are provided to take the end thrust of the engine shaft and of the vertical worm shaft, the latter having also a step and a top bearing. The windlass engine is located on the gun deck just forward of the windlass pocket and is of the two-cylinder, vertical, inverted, reversible type, having two 15-inch cylinders of 14-inch stroke.

Steering Engine.—The steering engine is of the Williamson Brothers' design, arranged for steam and hand control, the usual steam-steering wheels being fitted on the flying bridge, forward bridge, and in the conning tower. Electric and direct-acting indicators are installed showing the position of the helm, and an electric transmitter communicates to the hand-steering station. The engine is two-cylinder, horizontal, direct-

acting; diameter of cylinders, 15 inches; stroke, 12 inches; and diameter of piston rods $2\frac{1}{16}$ inches.

Electric Plant.—The electric plant of this vessel consists of two 100-kilowatt and three 50-kilowatt generating sets of 125 volts pressure at the terminals; about one thousand electric fixtures, complete, with all necessary incandescent lamps; twelve enclosed arc lamps of about 3 ampères capacity in engine andfirerooms; six 30-inch searchlights; two truck lights; two electric night-signaling sets, complete; two diving lanterns with 150-candlepower incandescent lamps; six portable electric ventilating sets; forty $\frac{1}{12}$-horsepower desk and bracket fans; six $\frac{1}{6}$-horsepower bracket fans; one electric whistle operator; all necessary panel and switch boards for lighting and power; with all wiring, accessories, fixtures, lamps, instruments, tools, spare parts and stores necessary for the proper manipulation, test and repair of the plant. The generating sets are secured to a common bedplate, the engine being direct-connected to the dynamo. The engines are of the vertical, cross-compound, General Electric Company's type, all working parts enclosed and lubricated under pressure, the introduction of oil into the cylinders being guarded against by a top plate on the enclosing covering, having soft-packed stuffing boxes for the rods and valve stems to work in. The dynamos are of the direct-current, compound-wound, multipolar type, and the wiring is arranged on the two-wire feed system.

50-kw. dynamo engines, number	3
Diameter of cylinders, H.P., inches	$7\frac{1}{2}$
L.P., inches	14
Stroke, inches	8
Diameter of piston rods, H.P., inches	$1\frac{3}{4}$
L.P., inches	$1\frac{3}{4}$
100-kw. dynamo engines, number	2
Diameter of cylinders, H.P., inches	10
L.P., inches	18
Stroke, inches	10
Diameter of piston rods, H.P., inches	$2\frac{1}{4}$
L.P., inches	$2\frac{1}{4}$

Motors.—In connection with the electric plant, and for the operation of the various auxiliaries outside the engine and

fireroom spaces (all of which, with the exception of the forced-draft blowers, ice machine, evaporator pumps, windlass and steering gear, are operated by electricity), there are installed the following motors:

Use.	No.	H P.		Winding	Current. (amp.)	Location.	Makers and remarks.
		Each.	Total.				
Ventilation.							
70-in. S. P. blower.	2	...	...	Shunt.	64	Protective deck.	B. F. Sturtevant Co
60-in. S. P. blower.	4	...	...	Shunt.	64	Eng. room and forward hold.	B. F. Sturtevant Co.
50-in. S. P. blower.	2	...	...	Shunt.	36	Berth and gun decks.	B. F. Sturtevant Co.
45-in. S. P. blower.	3	...	...	Shunt.	26	Prot., berth and gun decks.	B. F. Sturtevant Co.
35-in. S. P. blower.	6	...	...	Shunt.	15	Prot., berth and gun decks, power platform aft, and steering eng rooms.	B. F. Sturtevant Co.
No. 4 Monogram ..	6	...	...	Shunt.	10	Berth and gun decks and upper platform.	B. F. Sturtevant Co.
No. 3 Monogram...	3	...	...	Shunt.	6.80	Gun deck.	B. F. Sturtevant Co.
No. 1 Monogram...	1	...	...	Shunt.	1.96	Berth deck.	B. F. Sturtevant Co.
Boat cranes.							
CB 25.....	2	35	70	Series.	240	Boat cranes.	Gen. Electric Co.
CB 31.....	6	20	120	Series.	140	Bridge deck.	Gen. Electric Co.
Deck winches.							
Type M.....	7	25	175	*Series.	175	Main and upper deck.	Thresher Elec. Co.
Ammunition hoists...	27	2.5	67.5	Shunt.	19	Hold, amm'n passage and berth deck.	Gen. Electric Co.
Ammunition winches (whip hoists.)	3	3.5	10.5	Shunt.	26	Upper deck and bridges.	Gen. Electric Co.
Pumps, fresh water..	2	2	4	Shunt.	14.7	For'd hold and steering eng. platform.	Holtzer Cabot Co.
Sanitary.....	1	½	½	Shunt.	3.9	Engine room.	Gen. Electric Co.
General workshop....	1	10	10	Shunt.	30	Gen. workshop.	Gen. Electric Co.
Laundry.....	1	6	6	Shunt.	42	Laundry.	Gen. Electric Co.
Bread mixer.....	1	2	2	Shunt.	14	Bakery.	Gen. Electric Co.

*Shunt winding sufficient to control speed at no load.

VENTILATION.

All the spaces below the gun deck, and the officers' and crew's closets on the gun deck, are provided with artificial ventilation. There are fifteen systems (or units) of ventilation, and these are grouped into four methods, viz:

The Supply and Exhaust Method.—This method is applied to all compartments where it is difficult both to supply the fresh air and to exhaust the foul by natural convection. Under this head come the dynamo rooms and the steering-engine room.

The Supply Method.—Air is supplied by forced ventilation to all compartments below the gun deck and between the

'thwartship armor bulkheads, and also, where necessary, as in the magazines, exhaust ducts are provided to allow the escape of the foul air.

The Exhaust Method.—Where there is a natural inlet of fresh air through hatches and air ports, as in the case of living quarters aft on the berth deck, the foul air is exhausted by fans.

The Convection Method.—This is applied to the bunkers and firerooms where there is a natural flow of heated air upward through hatches and pipes provided for that purpose, and ventilators and supply pipes are arranged to carry the fresh air to supply the need.

The engine rooms are ventilated by four electric blowers, locted two on and two just below the protective deck. These blowers supply fresh air from ventilators to the four main ducts leading inboard and outboard in each engine room. From there ducts branches are led to the various points in the engine rooms where cool, fresh air is most needed. The preceding table of motors gives the number, type and location of the different ventilation blowers.

STANDARDIZATION OF SCREWS.

It having been decided to try the vessel according to the standardized screw method, progressive runs over the measured mile at Provincetown, Mass., were made and the data recorded in Tables I and II was obtained. From this data the curves on plates I and II were plotted.

It was determined that 143.081 revolutions per minute of the main engines would be required to give the contract speed of 22 knots per hour.

The standardization runs were commenced at 8·31 A. M. on June 28th and completed at 2·15 P. M. The draught and corresponding displacement at the beginning and end of the runs were as follows:

	Beginning.	*End.*
Draught, forward, feet and inches	22- 6	22-1
aft, feet and inches	22-10	22-6¼
mean, feet and inches	22- 8	22-3¾
Displacement, tons	9,800	9,620

The coal used was New River, run of the mine.

Table I.—U. S. S. *CHARLESTON.*

Data from Progressive Runs over Measured-Mile Course off Provincetown, June 28, 1905.

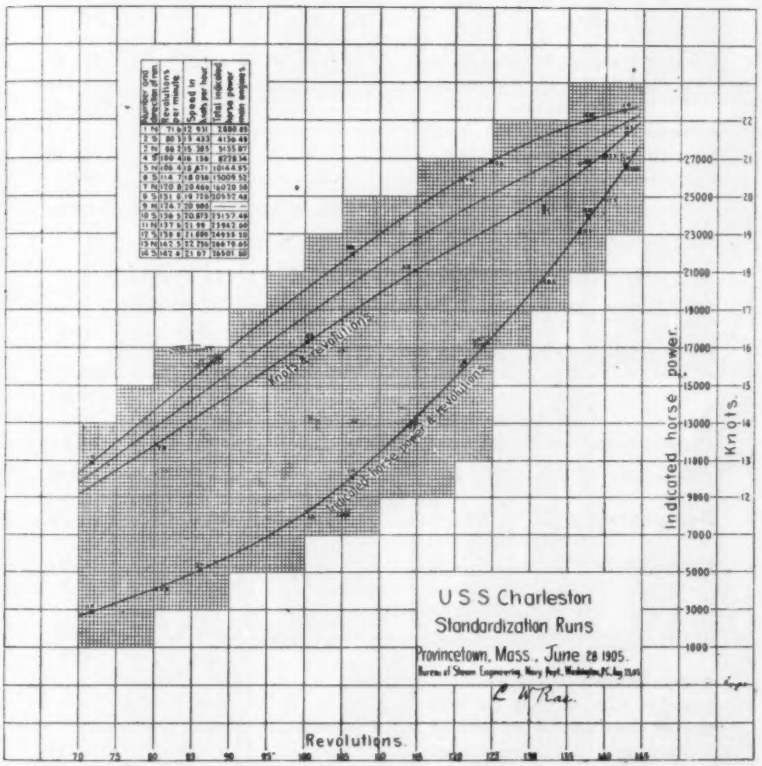
Number and direction of run.	Time on course.	Revolutions of main engines.					Speed in knots per hour.	Indicated horsepower.												
		To make one knot.		Per minute.				Starboard engine.					Port engine.					Total both main engines.	Auxiliaries in operation (estimated).	Total all machinery.
		Starboard.	Port.	Starboard.	Port.	Mean.		H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.	H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.			
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
	m. s.																			
1-N	4-38.4	333.3	331.3	71.8	71.4	71.6	12.931	583.37	472.2	*148	169.70	1,373.27	580.06	469.5	213.97	244.09	1,507.52	2,880.89	...	...
2-S	4-28	361.8	355.5	81	79.6	80.3	13.433	790.7	659.5	*247.78	259.82	1,957.8	1,023.8	590.6	275.43	288.86	2,178.69	4,136.49	...	...
3-N	3-54	337.7	339.6	86.6	87.1	86.2	15.385	1,045.7	859.7	*285.65	321.64	2,512.59	1,071.1	829	349.14	393.35	2,642.48	5,155.07	...	...
4-S	3-43.07	376	370.8	101.1	99.7	100 4	16.136	1,688.5	1,367.7	*490.8	563.24	4,110.24	1,609.4	1,342.9	542.9	622.95	4,118.1	8,228.34	...	...
5-N	3-14.93	348.1	343.1	107.1	105.6	106.4	18.471	2,066.8	1,702.1	542.42	786.57	5,097.83	2,037.8	1,663.4	561.5	784.4	5,047.1	10,144.93	...	...
6-S	3-19.6	382.8	380.4	115.1	114.3	114.7	18.036	2,441.95	2,068.9	1,190.2	1,000.35	6,701.8	2,438.6	1,987.1	844.5	1,037.5	6,307.72	13,009.52	...	...
7-N	2-55.9	357.7	350.5	122	119.5	120.8	20.466	2,960	2,674.3	965.96	1,503.52	8,103.88	2,860	2,564.3	1,200.8	1,291.6	7,916.7	16,020.58	...	...
8-S	3- 2.5	400.7	400.8	131.8	131.8	131.8	19.726	3,604.15	3,456.1	1,576.4	1,734.23	10,370.88	3,442.1	3,486.4	1,579.3	1,673.8	10,181.6	20,552.48	...	...
9-N	2-52.2	393.3	322.5	137	112.4	124.7	20.906	3,826.5	4,590.4	2,110.5	2,283.95	12,811.35	...	...	...	...	...	...	...	...
10-S	2-56.66	401.6	399.6	136.4	135.7	136.5	20.873	3,917.45	4,143.2	1,850.25	1,991.85	11,902.75	3,875.4	3,629.9	1,816	1,913.4	11,234.73	23,137.48	...	...
11-N	2-43.66	376	374.8	137.8	137.4	137.6	21.99	3,327.8	4,470.6	2,041.4	2,140.3	11,980.1	3,083.4	4,841.9	2,152	1,885.2	11,962.5	23,942.6	...	...
12-S	2-50.7	397	399.1	139.5	140.3	139.8	21.089	3,463.3	4,630.5	2,060.65	2,215.75	12,368.2	2,960.3	4,876.1	2,279.45	2,451.15	12,567	24,935.2	...	...
13-N	2-41.9	384.9	384.5	142.7	142.5	142.5	22.236	3,156.3	5,451.6	2,321.1	2,541.55	13,470.55	2,911	5,355.4	2,466.1	2,676.6	13,209.1	26,679.65	...	...
14-S	2-46.1	395.9	394.6	143	142.5	142.8	21.67	2,985.5	5,445.2	2,444.4	2,559.05	13,434.15	2,895.4	5,088.5	2,412.45	2,671.1	13,067.35	26,501.5	...	...

* Defective cards interpolated from port engine.

Table II.—U. S. S. *CHARLESTON.*

Data from Progressive Runs over Measured-Mile Course off Provincetown, Mass., June 28, 1905.

Number of run.	Time at beginning of run (approximate).	Which engine.	Steam pressures.			Mean effective pressures.				Cut off in cylinders from beginning.			Temperatures.							Revolutions or double strokes.								Air pressure, inches water.	Sanitary pump.
			In boilers above atmosphere.	At engine above atmosphere.	H.P. steam chest, gauge.	In first receiver, absolute.	In second receiver, absolute.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to low pressure.	Vacuum, inches.	H.P.	I.P.	L.P.	Outside air.	Engine-room working platform.	Fireroom working level.	Injection.	Discharge.	Hotwell.	Feed.	Circulating pump.	Main feed pumps.	Aux. feed pumps.	Hotwell pump.	Main air pump.	Bilge pump.	Aux. cond'r pump.	Fireroom blowers.		
1	...	S. P.	280	280	55	22	11.5	36	10.5	0	2.8	10.11	28.7	Full gear.			70	75	...	60	98	150	150	120	...	...	18	20	39	10	225	...	20
		P.	270	270	55	20	7.75	...	10.5	3.55	4.05	...	28.5					91	...	55	98	122	...	98	7	...	12	18	33	59	...	...	20
2	...	S.	290	290	73	27	13	43.25	13	.7	3.8	13.86	28.5				70	85	...	60	88	140	140	120	...	...	18	18	33	60	255	...	20
		P.	245	245	70	25	8	57	11.85	4.1	4.3	16.22	28.5					91	...	60	88	110	...	97	18	...	11	19	38	80	...	...	20
3	...	S.	254	97	33	15	...	53.5	16.85	1.85	4.4	16.16	29				68	86	...	60	94	128	130	116	...	...	18	20	39	60	270	...	20
		P.	248	96	34	12	...	54.5	15.2	4.75	5.35	17.98	28.25					98	...	60	92	98	...	95	18	...	13	19	44	60	...	...	20
4	...	S.	270	130	43	12	...	74	21.6	4.55	6.6	23.34	29.5				64	97	...	60	96	115	160	116	11.5	...	17	20	39	10	281	...	20
		P.	270	150	37.5	15.5	...	71.5	21.5	6.45	7.4	28.46	27.5					94	...	59	100	84	...	112	...	...	18	20	40	22	...	...	20
5	...	S.	292	157	58	19	...	85.5	25.4	6	8.7	28.21	28.7				64	86	...	58	102	92	150	116	...	...	18	20	39	10	295	...	20
		P.	290	162.5	47.5	18.5	...	85.5	25.15	6.3	8.8	28.31	26.75					88	...	58	118	85	...	114	15	...	13	18	38	31	...	...	20
6	...	S.	280	180	57	24	...	94	28.7	8.75	10.3	34.49	28.5				61	89	...	60	110	88	142	120	18.5	...	7	19	41	10	295	...	20
		P.	270	187	57.5	23.5	...	94.5	27.75	8.75	10.75	31.68	26.75					88	...	60	116	86	...	135	...	...	7	19	41	8	335	...	20
7	...	S.	295	224	65	29	...	107.5	35	11.9	12.8	39.23	25.75				70	84	...	58	109	90	138	122	...	...	20	20	41	44	...	...	18
		P.	272	215	65	27	...	106	34.25	11.9	12.8	39.23	25.75					84	...	58	109	90	138	122	...	...	20	20	40	8	375	...	18
8	...	S.	240	235	87	33	...	135.25	41.9	14.18	15.6	40.65	28				62	90	...	58	112	92	150	116	...	...	20	20	40	8			
		P.	240	237	83	33	...	115.75	42.25	14.2	15.05	45.78	25.50						...	58	118	90	...	147	15	...	18	21	38	38	6	380	...	18
9	...	S.	...	...	...	...	...	...	...	18.25	19.75	35.4	...				...	...	...	...	...	...	...	180	...	...	20	26	40	31	...	...	20
		P.	...	...	...	...	...	...	...	...	...	...	...						...	...	...	...	...	183	...	...	20	26	41	30	...	...	16
10	...	S.	250	247	108	38	...	197.25	48.5	16.7	17.3	31.6	28				66	88	...	60	102	98	150	120	...	...	20	23	40	6	495	...	16
		P.	250	247	93	36	...	126.5	42.7	15.85	16.7	49.03	26					92	...	59	112	89	178	180	...	...	22	21	44	38			
11	...	S.	250	109	41	...	...	107	51.8	17.55	18.4	31.5	27.5				71	88	...	60	120	100	160	120	...	...	20	24	40	6	505	...	16
		P.	255	252	96	37	...	99.4	56.25	18.55	16.25	46.56	27					91	...	59	116	110	...	171	27.5	...	23	21	37	39			
12	...	S.	252	112	41	...	...	110	53	17.5	18.8	32.52	27				64	87	...	60	120	104	155	124	...	...	24	23	44	6	500	...	16
		P.	255	252	110	42	...	93.5	55.5	19.25	20.7	33.07	25.5					91	...	59	118	112	...	169	28.5	...	18	21	41	34			
13	...	S.	255	197	45	...	...	98	61	19.27	21.1	35.87	27				64	86	...	60	122	102	162	200	...	...	24	24	48	6	540	...	16
		P.	260	257	115	44	...	90.5	60	20.5	22.25	35.74	26					92	...	58	118	112	...	173	30	...	29	19	43	30			
14	...	S.	250	250	120	45	...	92.5	66.8	20.25	21.2	35.65	27.5				68	86	...	60	120	100	155	200	...	...	24	24	48	6	445	...	16
		P.	260	257	115	44	...	90	57	20.05	22.2	34.3	26					91	...	58	118	111	...	178	32	...	36	21	72	34			


Number and direction of run.
Revolutions per minute.
Speed in knots per hour.
Total indicated horse power, mean engines.
Knots per hour.
Indicated horse power & revolutions.
Indicated horse power.
Knots.
Revolutions.
U S S Charleston
Standardization Runs
Provincetown, Mass., June 28 1905.
Bureau of Steam Engineering, Navy Dept. Washington, D.C. Aug 23, 05
L. W. Rae.

8 B 119

OFFICIAL FOUR-HOURS' TRIAL.

At 7·30 A. M. on June 29, the *Charleston* stood out to sea for the official four-hours' trial which commenced at 8·53 A.M. The weather was clear and pleasant with a light S.W. breeze and smooth sea.

The draught and displacement at the beginning of the trial were as follows:

Draught, forward, feet and inches	22- 3¼
aft, feet and inches	22-11½
mean, feet and inches	22- 7⅜
Allowance for anchor, etc., inch	⅜
Mean draught, feet and inches	22- 7¾
Displacement, tons	9,775

The estimated displacement at end of trial was 9,587.8 tons. A synopsis of the data obtained follows.

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

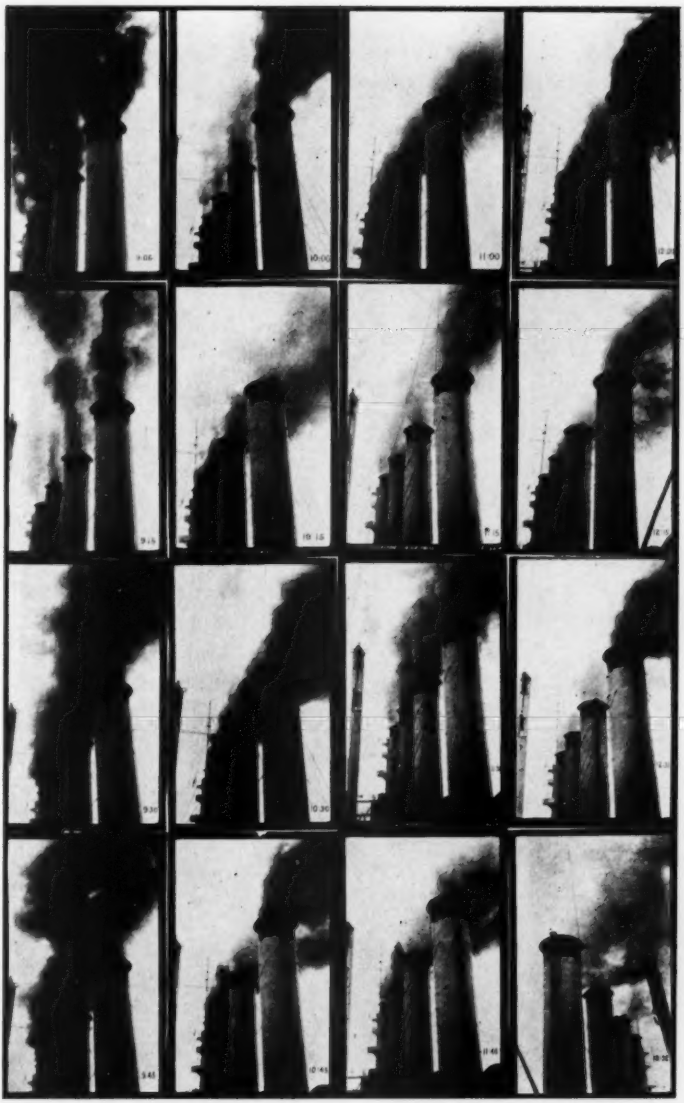
	Starboard.		*Port.*
Steam pressure at boilers (per gauge), pounds		286.7	
H.P. steam chest (gauge), pounds	252.8		254.2
1st receiver (absolute), pounds	125.7		122.9
2d receiver (absolute), pounds	47.4		45.4
Vacuum in condensers, inches of mercury	27.3		26.42

Revolutions, or Double Strokes, per Minute. (*Average of one-half hourly observations.*)

Main engines (mean for trial)	143.2		143.59
average for both		143.395	
Pumps, main air	24.8		23.9
circulating	192.2		181.8
feed (forward and after)		32.4	
hotwell	45.8		35.9
bilge	44.7		47.3
Blower engines (average)		529.5	
Speed of ship, in knots per hour		22.04	
Slip of propeller, in per cent. of its own speed, on mean pitch and average speed	23.96		24.16
Air pressure of firerooms, inches of water		2.48	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Average of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	91.463	83
I.P. cylinder	58.823	59.875
F.L.P. cylinder	21.954	21.438
A.L.P. cylinder	21.02	22.913
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	55.54	55.483



PHOTOGRAPHS OF SMOKE TAKEN DURING TRIAL, U. S. S. "CHARLESTON."

INDICATED HORSEPOWER.

	Starboard.	*Port.*
Main engines, H.P. cylinder	2,956.1	2,689.9
I.P. cylinder	5,275.6	5,384.5
F.L.P. cylinder	2,653.7	2,598.4
A.L.P. cylinder	2,540.8	2,777.1
Total	13,426.2	13,449.9
both engines	26,876.1	

AUXILIARIES.

Air pump, main	11.15	11.62
Circulating pumps, main	71.5	67.63
Feed pumps (main)	130.44	
Hotwell pumps	31.7	
Bilge pumps	7.44	
Auxiliary condenser pumps	6	
Blower engines (forced draft)	216	
Steering engine (estimated)	5	
Dynamo engine (one in operation)	72	
Total	630.48	
Main engines, air, circulating, hotwell and feed pumps	27,200.14	
Total all machinery in operation	27,506.58	

COAL.

Kind and quality	New River, hand-picked, excellent.
Pounds per hour	57,265

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	19.65
heating surface	.4298
Main engines, air, circulating, hotwell and feed pumps per square foot of grate surface	19.43
Main engines, air, circulating, hotwell and feed pumps per square foot of heating surface	.425
Pounds of coal per I.H.P. per hour (main engines, air, circulating, hotwell and feed pumps)	2.11
Pounds of coal per I.H.P. per hour, all machinery in operation	2.08
Pounds of coal per square foot of grate surface	40.9
heating surface	.895
Cooling surface (main condenser), square feet per (total) I.H.P.	.98
Heating surface, square feet per (total) I.H.P.	2.327

COAL CONSUMPTION OF U. S. CRUISERS *CINCINNATI*, *RALEIGH*, *NEW ORLEANS* AND *ALBANY*.

By Commander W. W. White, U. S. Navy (Retired), Member.

The accompanying table has been compiled from the results of a run, in squadron, of the *Cincinnati*, *Raleigh*, *New Orleans* and *Albany*, in December of 1903. A short time prior to the trip all ships had been docked and their bottoms cleaned. The coal used (Pocahontas) was from the same Government pile at Yokohama.

The principal data of these ships is as follows:

	Cincinnati and *Raleigh* (steel).	*New Orleans* and *Albany* (steel, wood-sheathed).
HULL.		
Length on L. W. L. (normal), feet	300	346
Extreme breadth, feet and inches	42	43– 9
Mean draught (normal), feet	18	18
Displacement (normal), tons	3,213	3,769
MACHINERY.		
Engines	4-cylinder, triple-expansion.	3-cylinder, triple-expansion.
Diameter of cylinders, inches	24 × 44½ × 57 × 57	31 × 46 × 70
Stroke, inches	33	30
Boilers	8, B. & W.	4, double-ended (Scotch).
Designed pressure	225	160
Grate surface, total, square feet	506	474
Heating surface, square feet	21,016	13,768

The auxiliary machinery in operation for all ships during the run was very closely the same, with the following exceptions: On the *Cincinnati*, all fresh water needed both for ship's use, and to supply deficiency of boiler feed, was made, as required, by the evaporating plant. Other ships distilled water for ship's use, but to replenish feed water drew frequently

UNITED STATES ASIATIC FLEET-CRUISER SQUADRON.

Performance data, Yokohama, Japan, to Honolulu, H. I. = 14 days' run. Data recorded at 10 A. M. daily.

December 3–16, 1903.	Date.	4	5	6	7	8	9	10	11	12	†12	13	14	15	16	Total.
Distance made good, knots.	* By observation..	‡225.3	224.3	‡227.8	228.7	244	232.8	241.8	252	245.8	‡242.3	‡292.4	310.3	301.3	‡289 5	3,558.3
	* By log.....	227.6	229.6	230.1	225.1	248.3	241.4	242.2	244.2	245.8	244.7	295.4	303.3	311.1	292.4	3,581.2
	Average per hr.	9.4	9.7	9.6	9.6	10 3	9.8	10.2	10.6	10.3	10.2	12.3	13	12.7	12.2	...
Average revolutions per minute.	*Cincinnati*.....	70.2	68.8	70.1	68.4	73.9	73.4	73.7	73.8	74.3	74.4	89	92.5	94	90	...
	Raleigh	72.9	72.4	75.8	72.6	78.6	78.7	78.6	78.6	79.7	79.5	96	99.4	101.4	97.8	...
	New Orleans....	73	71.5	73.7	72	76.5	76.9	77	77	76.4	77	91.6	95	97.2	94.2	...
	Albany.....	74.7	74.6	77.2	74.9	79.5	77.9	79.9	79.3	79.7	79.8	95.2	98.3	100.5	97.3	...
Coal per day, tons.	*Cincinnati*.....	29.2	29.4	29.4	28.1	31.9	35.7	31	30.3	30.5	27.8	34.7	34.1	37.4	32.5	442
	Raleigh	36	35.3	28	24.6	30.2	28.8	30.1	31.6	30.5	30.6	45.3	48.6	50.6	52.4	502.6
	New Orleans....	30.5	31.7	32.8	31.8	32.9	33.2	34 5	35.1	32.6	33.1	44.2	44.7	47.8	47.7	512.6
	Albany.....	37.6	35.8	35.4	33.4	34.2	35.2	35.4	34.7	35.6	35	43.9	47.4	50.5	54.9	549

		Cincinnati.	*Raleigh.*	*New Orleans.*	*Albany.*
Draught, feet, inches, and Displacement, tons.	December 3.				
	Forward.....	19	18	18– 6	19– 4
	Aft.....	20– 6	20	19– 6	19–10
	Displacement ...	3,655	3,450	4,030	4,190
	December 16.				
	Forward.....	15– 7	14– 6	15– 6	15– 6
	Aft.....	20	18– 6	18– 2	17– 6
	Displacement ...	3,180	2,855	3,420	3,330

* Average of all ships.

† Crossed 180 degrees meridian.

‡ Computed from average of patent logs. (Course changed.)

Coal, Pocahontas from Government pile, Yokohama.

Ships' bottom clean.

Sea, moderate, 11 days; smooth, 3 days.

upon reserve feed stored in double-bottom compartments. Main air pumps of the *Cincinnati* and *Raleigh* are independent, while those on the *New Orleans* and *Albany* are attached to and worked by beams from the main engines; an exhaust feed-water heater is fitted on the *Cincinnati* and *Raleigh*, but there is no such device on the other two ships.

From an inspection of the table it will be observed that during the first ten days of the run there was but little variation in speed, the average being 9.97 knots per hour. The following is a recapitulation of the important data during that time:

	Cincinnati.	*Raleigh.*	*New Orleans.*	*Albany.*
Average revolutions	72.1	76.7	75.1	77.8
steam pressure, boiler (gauge)	193	167.3	127.7	132.3
steam pressure, engine (gauge)	188	162.3	125.7	127.3
steam pressure, 1st receiver (absolute)	34.1	33.8	21.7	22.3
steam pressure, 2d receiver (absolute)	14.3	17.4	13.3	8.9
Vacuum	22.2	21.9	22.7	23.3
Temperature of feed (Fahrenheit)	199.2	180	111.5	124.1
Water distilled, per day, gallons	6,662	*...	3,055	2,400
I.H.P., (main engines only)	897	867	801	808
Coal per hour, pounds	2,831	2,853	3,063	3,288
per I.H.P. (main engines only), lbs	3.16	3.29	3.82	4.07
square foot grate, per hour, pounds	11.19	11.28	12.93	13.87

*Not recorded.

BALANCE CYLINDERS.

By LUTHER D. LOVEKIN, ASSOCIATE.

[The following additional information on the subject of Balance Cylinders is in continuation of the article which appeared in Volume XVI, page 896, of the JOURNAL.]

For the benefit of those who may become interested in the balancing of valve gears I have taken the liberty of appending a recent extract from the "Marine Engineer," of London, England, under date of April 1, 1905, which is in answer to an article on "Slide Valve Resistances," by Mr. J. E. Cooper, in this same paper, under date of May 1, 1904, describing the Assistant Cylinders as fitted on the Royal Mail Steamer *Ophir*. These were Joy's Patent Assistant Cylinders, and give the reader a good opportunity for comparing the same with "The Improved Lovekin-Thom Assistant Cylinders," described previously, and as fitted on the Pacific Mail Steamer *Manchuria*. In order to determine the value of the Lovekin-Thom Assistant Cylinders under varying revolutions, I had several indicator cards taken while on the trial of the *Manchuria*, and made a direct comparison with the ordinary balance cylinder commonly employed under these same conditions. All cards have been plotted fully, and show at a glance *the great superiority of a properly designed assistant cylinder over the old form of balance cylinder, under all possible conditions*, such as varying revolutions, varying receiver pressure, etc., etc.

It will be noticed that with the use of the ordinary balance cylinder the *maximum pressure per square inch on all parts of the valve gear never falls below 51 pounds*, while with the use of the Improved Lovekin-Thom Cylinder we have as low as *11.3 pounds pressure per square inch*, or, approximately, only one-fourth that of the ordinary balance cylinder. Thus

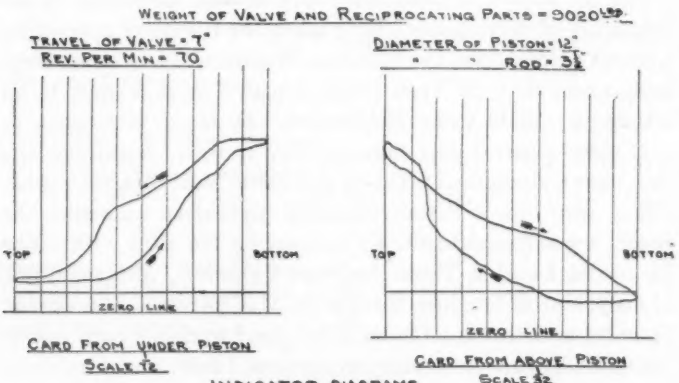
it will be seen that there is *everything to gain and nothing to lose by its general adoption on all reciprocating valve gears.*

Further attention is directed to the relative size and weight of these two cylinders, which are clearly shown on this same sheet.

SLIDE VALVE RESISTANCE.

To the Editor of the "Marine Engineer:"

Dear Sir: I thank you for the data supplied in response to my request published in your January issue. I desire also to thank Mr. Jasper E. Cooper for his courtesy in sending me



INDICATOR DIAGRAMS FROM JOY'S ASSISTANT CYLINDER.

FIG. I.

the indicator cards of the Joy Assistant Cylinder, which correspond with the data mentioned.

I append herewith the cards and the diagrams which I have plotted therefrom. I will explain all the steps thoroughly, so that they may be easily followed and verified:

First, divide the cards into a convenient number of parts, say ten, as would be done in the case of any indicator card to be worked up by hand. Draw a line below the atmospheric line to indicate absolute zero, and from this measure the ab-

solute pressures at each point in the stroke, both up and down, as given by the card. Next multiply the pressures thus obtained by the area of the piston on which they act. Thus, the pressures measured from the card taken from the top of the piston will be multiplied by 113.1 square inches, this being the area of a piston 12 inches in diameter. For the bottom card multiply by 103.48 square inches, as this is the effective area after deducting the area of the rod.

We now have the absolute total pressures of steam acting above and below the piston at all points of the stroke. These are the forces which the steam sets up, one opposing the other. There is another known force, that of gravity, which acts constantly downward with a force equal to the weight of the valve and gear. In this case the weight is 9,020 pounds. It is evident that this force acts with the steam pressure on top of the piston, so these two must be added together to give the total downward force. The upward force consists solely of the pressure of the steam below the piston.

Proceeding now to lay out these forces in graphic form, we refer them to a common base line, as shown in Fig. 2. The pressure below the piston is plotted direct from each corresponding point in the indicator card. Of course, this is from the card from below the piston, and care must be taken to distinguish between the up and down strokes.

In order to plot the downward pressures, first take that due to gravity, or 9,020 pounds, and lay that off on a parallel line the proper distance above the base line. This represents the constant downward pressure due to the weight of the gear. In order to get the total downward pressure we must add that due to the steam on top of the piston. This is done in exactly the same manner as for the bottom steam pressure, except that it begins at the gravity line instead of at the base line.

We now have the upward and downward forces laid down so that they may be analyzed and compared. A little study of the indicator cards will show that the lines correspond properly, and any point in question can be verified by working up the data given.

The curve on the diagram marked "Resultant" is obtained from the two curves of upward and downward forces by taking their differences at each ordinate, and measuring off that distance from the base line. Thus it will be noticed that in general the total downward forces exceed the upward forces. This is brought out clearly in the "Resultant" curve, which shows that only for a small portion of the stroke near the bottom does the line cross the base line, and indicate a resultant upward force on the eccentric strap. This resultant curve is easily verified by stepping off the vertical distance between the curves of upward and downward forces, and measuring that distance up or down from the base line according as the upward or downward forces predominate.

Thus far the analysis has been very simple, merely a little arithmetic and common-sense interpretation of the indicator diagrams. We must now introduce the force of inertia, which is not generally so well understood. Yet I trust I may put it so that all may understand its action in this case. We know that a stationary body requires force to overcome its inertia and start it moving, and a moving body requires force to overcome its inertia and stop it. This is represented in the diagram by the diagonal line crossing the base line at the middle of the stroke. At 70 revolutions per minute it takes a force of over 4,000 pounds to start the valve and gear downward, and the same force to stop it at the end of the stroke, and start it up again. At the middle of the stroke the force of inertia becomes nothing, for here it changes from one direction to the other. It has been retarding the acceleration of the valve and gear up to this point, but now begins to resist the slowing down that occurs during the last half of the stroke. We leave the weight of the gear out of the discussion, after having taken care of it as previously shown by including it as a fixed portion of the downward forces.

In order to locate the curve or line which indicates the force of inertia at each point in the stroke we must have recourse to higher mathematics. The expression for the valve of the inertia force at any point of the stroke is

$$F = \frac{r\pi^2 N^2 W}{900\ G}\left(\cos\theta + \frac{\cos 2\theta}{n}\right)$$

where r = radius of eccentricity, in feet,
N = revolutions per minute,
W = weight of valve and reciprocating parts, in pounds,
G = force of gravity, equals 32.16,
θ = angle made by the eccentric with the line of travel,
n = ratio of length of eccentric rod to radius of eccentric, equals $\frac{r}{l}$.

(Refer to Fig. 3.)

The proof of this expression is so long that I will not attempt it here, except to show that it is equivalent to the expression "$rw^2 \cos\theta \times$ Mass," which Mr. Cooper used in his article.

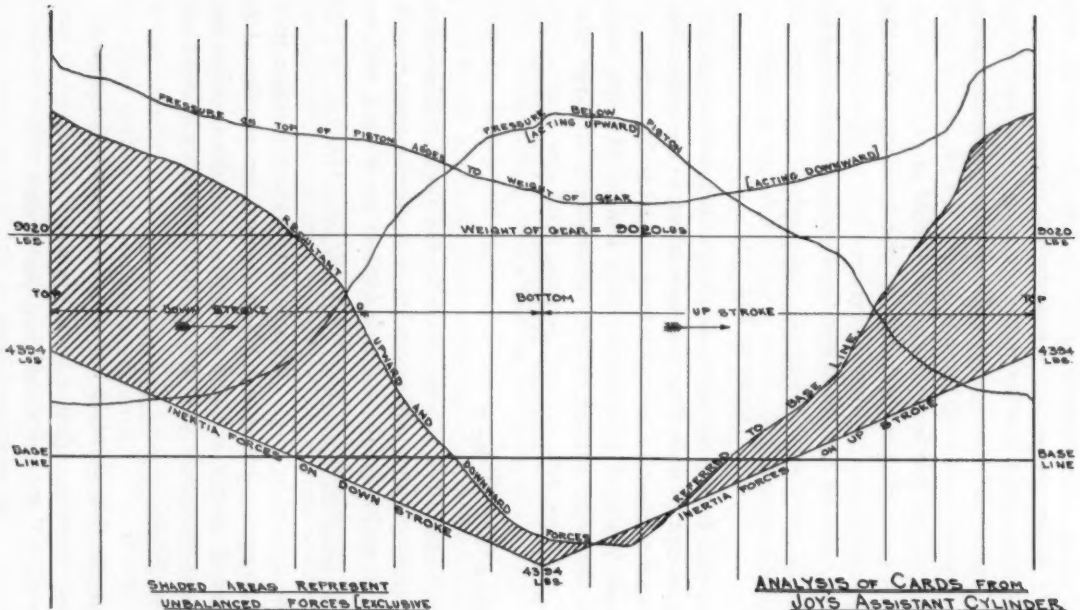
For all practical purposes we may assume that the ratio "n" of the eccentric rod to the eccentricity is so large that the fraction $\frac{\cos 2\theta}{n}$ becomes negligible. This leaves the expression $\frac{r\pi^2 N^2 W}{900\ G}\cos\theta$.

Now if instead of $2\pi N$, I take w, the angular velocity as taken by Mr. Cooper, and instead of N revolutions per minute I take 60 multiplied by "n" revolutions per second, and, instead of mass, its equivalent $\frac{W}{G}$, the fraction reduces to $r\,w^2 \cos\theta$ multiplied by mass. This is exactly the expression used by Mr. Cooper in his article. Now the maximum value of F, the force of inertia, is found when $\theta = 0$ degrees, or when the valve is at the end of its stroke. Cos θ then becomes unity and $F = \frac{r\,\pi^2\,N^2\,W}{900\ G}$ or $F = .0003408\ r\ N^2\ W$.

From the data furnished by Mr. Cooper with the cards

$r = 3\frac{1}{2}$ inches $= .292$ feet,
$N = 70$ revolutions per minute,
$W = 9{,}020$ pounds.

Hence $F = 4{,}394$ pounds.



SHADED AREAS REPRESENT UNBALANCED FORCES [EXCLUSIVE OF FRICTION] WHICH HAVE TO BE OVERCOME BY THE ECCENTRIC

FIG 2

ANALYSIS OF CARDS FROM JOYS ASSISTANT CYLINDER.

FEB. 4-05 ROBT. S. RILEY.

VERTICAL SCALE 4000 = 1 INCH.

Under the assumption that $\frac{l}{r}$ is infinitely large, the line indicating the inertia force is a straight line cutting the base line at its center and reaching the value of 4,394 pounds at each end of the stroke. We therefore lay off this line in the manner described in the diagram.

The ideal condition is attained when the resultant of gravity and the steam forces coincide with this inertia curve, for then all forces are exactly balanced.

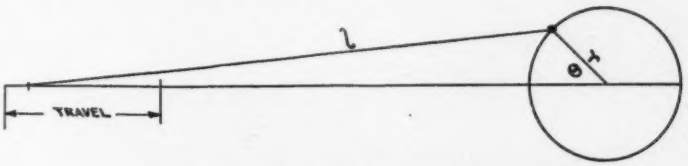


DIAGRAM TO ILLUSTRATE FORMULA FOR INERTIA FORCES

FIG. 3.

How far the Joy Assistant Cylinder fails to attain this ideal condition is shown by the shaded area between the two lines. Thus, on the first half of the down stroke and the last half of the up stroke there is an unbalanced downward pressure of nearly 10,000 pounds. This is more than the weight of the gear, showing that the assistant cylinder not only fails to relieve the stress on the eccentric, but actually adds to it.

I leave this without further comment to the consideration of your readers. I should be glad to hear from Mr. Cooper, or anyone else who can refute the facts shown up by this method of analysis.

Of course it is understood that I leave friction out of the discussion, just as Mr. Cooper did in his analysis. Could we obtain a satisfactory value for that force we might change the unbalanced forces materially, making them less on the down stroke and greater still on the up stroke.

The fact is that the principle of admitting and exhausting steam, as done in the Joy Cylinder, is wrong. It often stops vibration, but that is because it imposes a more or less steady load as compared with no balance cylinder, or one of the ordinary type. The problem must be solved by a somewhat different action of the steam used. It can never be quite perfect, of course, although it is nearly so in a different type of assistant cylinder. I refer to the Lovekin-Thom improved type, as recently developed in America. But that is another story.

Thanking you, sir, for the space allowed me, and trusting that I may hear from anyone who fails to understand my analysis, I remain, yours truly,

ROBERT S. RILEY,
(*B. Sc., Extra First-Class Engineer.*)
236 West Logan Square, Philadelphia, Pa.
March 6, 1905.

THE LOVEKIN DIFFERENTIAL REGULATING VALVE FOR SUPPLYING STEAM TO A SERIES OF ASSISTANT CYLINDERS FROM THE H.P. RECEIVER.

It has been observed from actual practice that the receiver pressure in an engine varies approximately as the square of the revolutions. This is exactly the relation that is desired in the Lovekin-Thom Assistant Cylinders, viz: the pressure should vary as the inertia. Hence an ordinary reducing valve for keeping a constant pressure would be unsuitable except at one particular speed. On the other hand, the differential reducing valve is suitable at all speeds, for it will be seen upon examination that the pressure delivered by the valve bears a constant ratio to the receiver pressure from which the supply is taken. This ratio is fixed by the relative areas of the differential piston which operates the valve. It is much simpler than any other type of reducing valve; there is only one moving part, and nothing to get out of order.

The great advantage in using a valve of this type is that it enables us to reduce both the weight and the cost of the

Advantages Actually Shown By The Lovekin-Thom Assistant Cylinder as applied to SS Manchuria.

1- Used high pressure steam, thereby cutting down size and weight of cylinder to a minimum.

2- Takes into consideration both the weight of the valve gear and its inertia. The latter at full speed is the most important force to be overcome.

3- The steam being taken from a receiver varies nearly as required by the inertia, which varies as the square of the revolutions per minute. With the ordinary balance cylinder only one receiver pressure is correct and that can only balance any other force than gravity, and can not balance this properly except at the one speed that gives the required receiver pressure. — See diagrams below.

4- At usual speeds the balance is practically perfect, and there is no work done by the eccentric and crank shaft. — See diagrams from actual indicator cards on trial trip.

5- Wear and tear of valve gear is eliminated; hence no expensive overhauling and repairs. It also required less attention while running.

6- No vibration and racking of engine due to the great weight and inertia of the valve and gear. It weighs over 8000 lbs. and has a travel of 11".

7- No change in length of valve gear by wearing down, hence the setting of valve remains perfect and the economy of the engine is maintained.

8- Saving in the use of oil, as there is no tendency for the strap to heat up under such light load.

9- Saving in I.H.P. which is otherwise needed to overcome friction of valve gear.

10- Absolute safety against breakdown of valve gear as it has a great margin of strength over the work now required of it. With the Lovekin-Thom Cylinder, the gear might be safely cut down 25% of the usual weight.

Note: The diagrams for the Lovekin-Thom Assistant Cylinder were plotted from indicator cards taken on trial, with inertia calculated for the speeds corresponding.

The diagrams for the ordinary type of balance cylinder are calculated from actual L.P. receiver pressures with inertia corresponding.

S.S Manch

Cards taken from L.P. Assistant

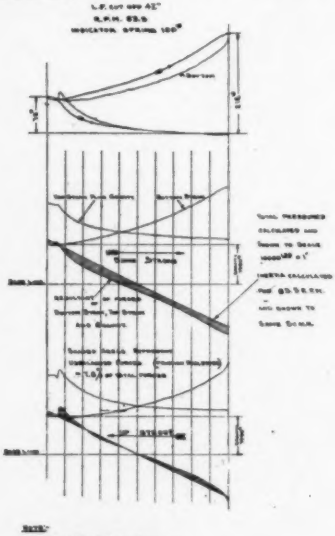
Diameter of Cylinder 11"

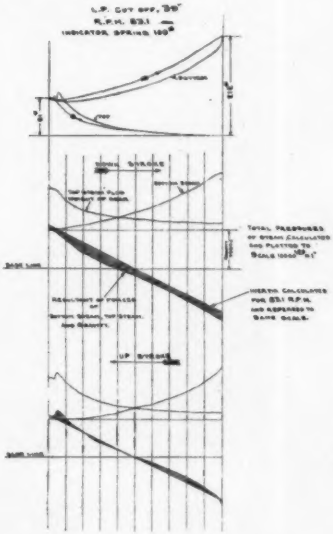
Diameter of Rod 3"

Projected Area of Eccen

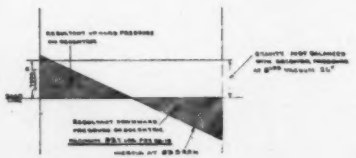
Travel of Va

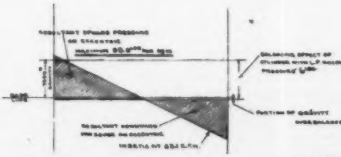
Weight of Reciprocating


L.P. cut off 42"
R.P.M. 83.5
Indicator Spring 100
Total Pressures calculated and drawn to Scale 1000 = 1"
Inertia calculated for 83.5 R.P.M. and drawn to same Scale.
Resultant of Forces of Steam, Top Steam and Gravity.
Up Stroke
Down Stroke


L.P. Cut off, 39"
R.P.M. 83.1
Indicator Spring 100
Total Pressures of Steam calculated and plotted to Scale 1000 = 1"
Inertia calculated for 83.1 R.P.M. and appeared to same Scale.
Resultant of Forces of Steam, Top Steam and Gravity.
Up Stroke
Down Stroke

Diagrams below show the forces developed by an ordinary balance cylinder, de signe
condenser absolute pressure. Diameter of cylinder 23¾" - Area = 443. sq. in: total. pr





Shaded areas represent unbalance

MANCHURIA

P ASSISTANT CYLINDER PORT ENGINE - 21-AP-04

INDER 11" AREA ON TOP OF PISTON 95.03 SQ.IN

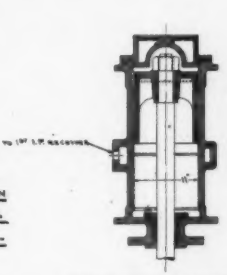
OD 3" AREA OF ROD 7.06 " "

AREA ON BOTTOM OF PISTON 87.97 " "

EA OF ECCENTRIC = 168.7 SQ.IN.

TRAVEL OF VALVE 11.5"

ECIPRICATING PARTS OF VALVE GEAR 7930 LBS.

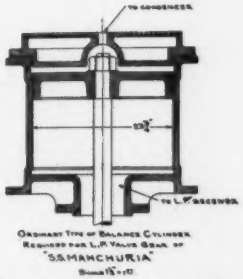


LOVEKIN-THOM ASSISTANT CYLINDER AS FITTED ON S.S. MANCHURIA FOR L.P. VALVE GEAR

SCALE 1½"=1'0"

TOTAL WEIGHT (WITHOUT VALVE STEM EXTENSION)

735 LBS.

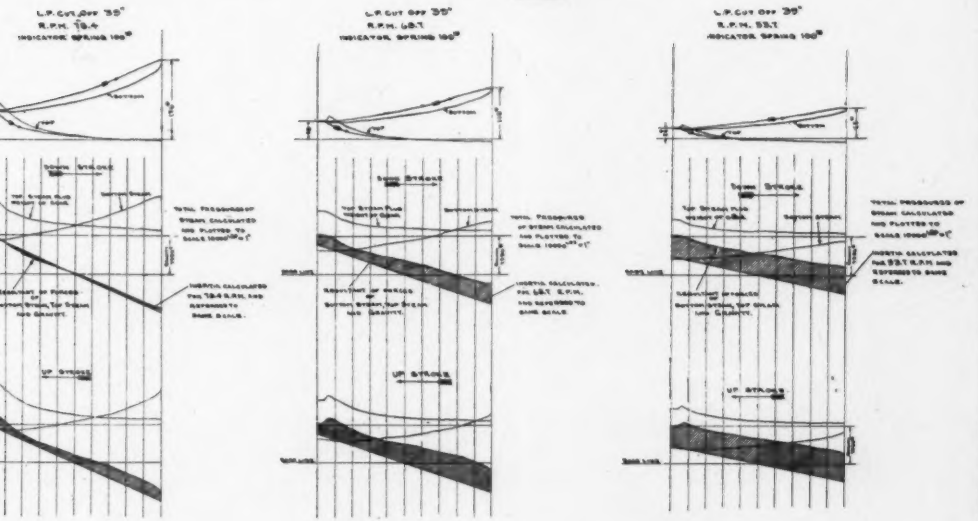


ORDINARY TYPE OF BALANCE CYLINDER REQUIRED FOR L.P. VALVE GEAR OF "S.S. MANCHURIA"

SCALE 1½"=1'0"

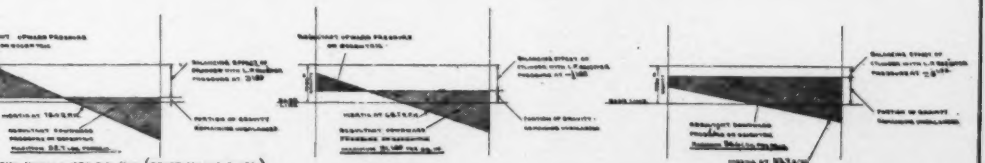
TOTAL WEIGHT (WITHOUT VALVE STEM EXTENSION)

1740 LBS.



ER, DESIGNED TO BALANCE GRAVITY ONLY, WITH L.P. RECEIVER STEAM AT 6 LBS. GUAGE = 18 LBS. ABOVE

N: TOTAL PRESSURES CALCULATED AND REFERRED TO SAME SCALE AS ABOVE 10000 LBS. = 1 INCH.



NT UNBALANCED PISTON (FRICTION NEGLECTED.)

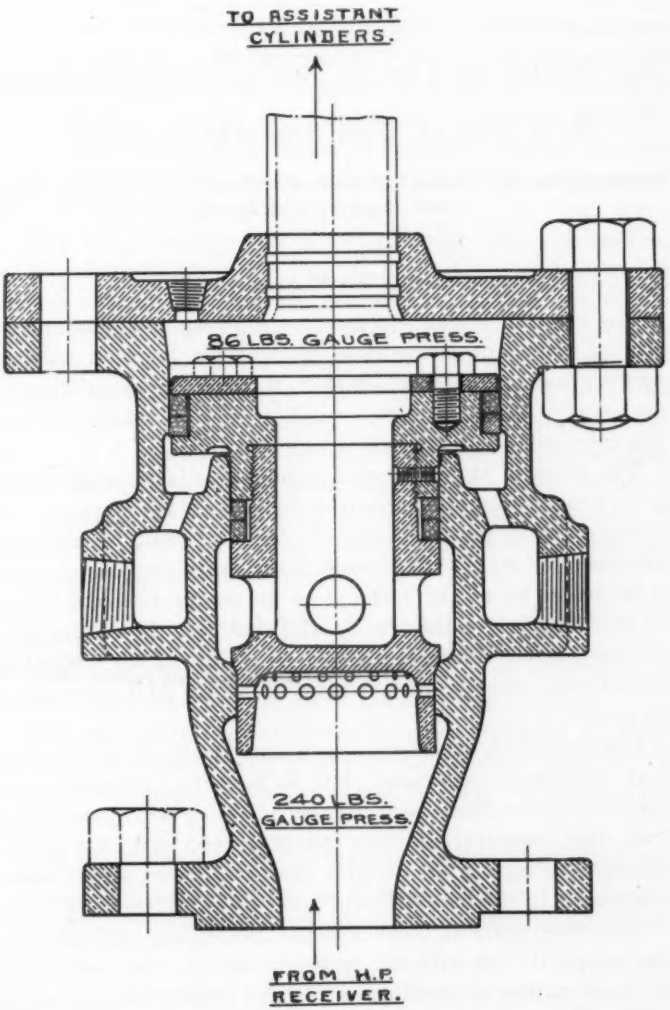
assistant cylinders themselves where we have a series of cylinders of the same size and requiring the same pressure to each one. This is particularly noticeable in the cases of the cruiser *Washington* and the battleships *Kansas* and *New Hampshire* of the United States Navy. Each of these vessels have two four-cylinder triple-expansion engines, each engine having seven piston valves. The main parts of the valve gear, viz: eccentrics, eccentric straps, eccentric rods, links and blocks, valve rods and suspension rods, are interchangeable, and even the valves are of the same size on the intermediate- and low-pressure engines. Hence, we have practically the same inertia and gravity to overcome in each case, which means that all the assistant cylinders may be of the same diameter. It will be very apparent to all engineers that the cost of patterns will be greatly reduced under these conditions.

This regulating valve can be placed on the high-pressure receiver, which will necessarily give the highest and also the most definite pressure for maximum running conditions of the engine. Where there is only one cylinder to be supplied, such as in the case of the *Manchuria* low-pressure valve gear, all we need is a pipe led from the most suitable receiver direct to the assistant cylinder. This pressure is chosen while the diagram is being made, so as to give the best possible results, and in the case of the *Manchuria* it was found that the first intermediate receiver was the most suitable.

With the sleeve type of assistant cylinder as fitted on the Steamship *Texan* and also the second M.P. valve gear of the *Manchuria*, no pipes whatever were required. Each cylinder was supplied with steam from its own receiver, and was placed directly over the valve chest in a similar manner to the ordinary balance cylinder. It is very important to know that in no case can an assistant cylinder of proper design be placed on a valve gear, without having first worked out the necessary diagram for the particular case in hand, and no decision can be given as to which form of assistant cylinder is best suited to

meet the ever-varying requirements, until such diagram has been accurately made and all the conditions fully considered.

In conclusion, I would state that no engine builder or designer would consider for a moment the running of a reciprocating engine without having first made *provision for proper compression at the top and bottom of the stroke.* Why, then, should not a valve gear receive its proper consideration *and thus save the engineer many worries at sea*, without mentioning the expense repairs, such as *relining the eccentric sheaves with white metal*, and the continual adjustment of the valve gear, all of which can be corrected by the proper use of these cylinders?



THE LOVEKIN DIFFERENTIAL REGULATING VALVE FOR SUPPLYING STEAM TO A SERIES OF ASSISTANT CYLINDERS FROM THE H.P. RECEIVER.

THE VALUE OF THE FLAME OF COMBUSTIBLES.

By P. Mahler, Civil Engineer of Mines.

Translated from the "Revue Universelle des Mines," by B. F. Isherwood, Chief Engineer, U. S. Navy.

I.—VALUE OF FLAME COMBUSTIBLES.

The calorific power and the chemical compositions are, in general, sufficient elements of comparison between natural combustibles. These data permit, notably, the calculation of the value of flames, and the question may be asked whether this calculation has any practical utility.

The value of the flame of a combustible is the same thing as its temperature of combustion under constant pressure. It is measured by the thermometric degrees through which the gaseous products of the combustion are raised, supposing them to be heated by all the heat due to the combustion and solely by that heat. It must also be admitted that the combustion is complete and that it is realized by the aid of air under the atmospheric pressure, always nearly constant, as it takes place on grates.

The preceding definition is in accord as much as possible with industrial conditions; but, in fact, the temperature of combustion is a theoretical number, defining with precision a limit that practical industry cannot exceed with any given combustible and for the given bodies to be heated, and it seems interesting to compare different coals from this point of view.

The calorimetrical bomb and the elementary analysis having supplied* me with the necessary data for the calculation of temperatures of combustion, I have determined the value of the flames of a complete series of combustibles.

*"Etude sur les Combustibles," by P. Mahler, 1903, published by Beranger.

II.—DETERMINATION OF THE TEMPERATURE OF COMBUSTION.

The determination of the temperature of combustion is done without difficulty, and the method I have followed is an old one. Nevertheless, I will cite the principal points, because they can be of service to engineers in many cases, as in the examination of the functioning of furnaces,* and of the economic results of gas motors calculated from the temperature of detonation of an explosive, etc.

Be q the calorific power of unit of weight of a combustible under constant pressure. Be C_m the mean specific heat† between O and T, absolute temperature of any one of the gases whatever composing the gases of combustion; let it be taken in relation to the weight of the gas which occupies a volume equal to 22.3 liters. Be N the number of molecules (22.3 liters) of this gas resulting from the combustion. As the energy of a gaseous mass depends wholly on its temperature, and as the coal in question is burned at a temperature very near 0° C. or T_0, the heating due to the combustion between T⁰ and T_1 is, according to our hypothesis, united closely to the calorific power by the expression

$$q = \Sigma N(C_m T_1 - C_m T_0),$$

Σ signifying that the gases of combustion are a mixture of several gases for which N and C_m have special values.

The analysis of the combustible gives N, and if C_m were constant, we could easily deduce from the above equation T_1, since the temperature sought for $\theta = T_1 - 273°$ in Centrigrade degrees.

But the problem is more complex. Messieurs Millard and

* "Le Chauffage Industriel," by Mr. E. Damour.

† dq being the elementary heating of 1 kilogramme, the true specific heat at the temperature t is

$$\frac{dq}{dt} = c$$

by definition.

Between the true heat and the mean heat from o to t, we have, in general, the relation:

$$S_o^t c\,dt = C_m t, \quad c = \frac{dC_m t}{dt}.$$

Le Chatelier have demonstrated that the specific heat of the gases of combustion increase with their temperature, this phenomenon becoming very sensible at high temperatures. According to these physicists it should be represented by a function

$$C_m = \alpha + \beta T.$$

The calculation of T_1 and, consequently, of θ, depends then on an equation of the second degree:

$$q = \Sigma N [(\alpha + \beta T_1) T_1 - (\alpha + \beta T_0) T_0]$$
$$q = \Sigma N [\alpha (T_1 - T_0) + \beta (T_1^2 - T_0^2)]. \qquad (1)$$

Messieurs Mallard and Le Chatelier have deduced from their experiments the values of α and of β relative to the different gases.*

I ought to say that in these later days certain very exclusive partisans of the law of Clausius regarding specific heats have contested the law of increase shown by Messieurs Berthelot and Vieille and by Messieurs Mallard and Le Chatelier. The law of increase, nevertheless, does not for this reason appear any less conformable to the truth. And if we reflect on the considerable difference existing between the temperatures of combustion calculated according to this law and the highest temperatures obtained in the best industrial furnaces (Martin-Siemens' furnace, 1,600°), we may even suppose—for the effect of the dissociation is feeble—that the increases resulting from the known experiments are inferior to the truth.

Under any circumstances the exactness of the experiments of Messieurs Mallard and Le Chatelier certainly surpass the precision I need for the comparative investigation that is the subject of this note.

Which being premised, we will now return to the calcula-

* Researches, experimental and theoretical, on the combustion of explosive gaseous mixtures; third memoir on the temperatures of combustion and the specific heats of gases at high temperatures, by E. Mallard and H. Le Chatelier ("Annales des Mines," 1885).

Report on the study of the questions relative to the use of explosives in presence of grisou (fire damp); temperature of detonation; by E. Mallard ("Annales des Mines," 1888).

tion of the temperature of combustion of coals and of other natural combustibles.

The gases of combustion comprise in general only perfect gases such as azote, and incidentally the oxide of carbon, and the condensed gases steam and carbonic acid. Relatively to the perfect gases, the remark must be made that their specific heats are nearly equal at all temperatures between the ordinary atmospheric temperatures and the temperature 3,000°.

According to the above physicists the following are the coefficients for the above gases:*

Perfect gases	$\begin{cases} \alpha = 6.5 \times 1{,}000^{-1} \\ \beta = 0.6 \times 1{,}000^{-2} \end{cases}$
Superheated steam	$\begin{cases} \alpha = 6.5 \times 1{,}000^{-1} \\ \beta = 2.9 \times 1{,}000^{-2} \end{cases}$
Carbonic acid	$\begin{cases} \alpha = 6.5 \times 1{,}000^{-1} \\ \beta = 3.7 \times 1{,}000^{-2} \end{cases}$

From these result the following expressions of the heating of a molecule of the above gases from T_0 to T_1 in large calories.†

* "Cours de Chimie Industrielle," by H. Le Chatelier.

† There may be needed the expression of the heating of a constant volume; for example, for the calculation of the temperatures of detonation, and also for the study of gas motors.

In this case, the coefficients of $[T_1^2 - T_0^2]$ are the same as above, and the coefficient of $[T_1 - T_0]$ is solely

$$4.5\,\frac{T_1 - T_0}{100}.$$

There may be also need of the heating of a molecule of methane, as follows:

$$4.5\,\frac{T_1 - T_0}{1{,}000} + 6\,\frac{T_1^2 - T_0^2}{1{,}000}$$

at constant volume.

The difference between the heating at constant volume results from the difference between the two specific heats [Cp, Cv] a difference due to the quantity of heat absorbed by the work of dilation accompanying the increase of temperature.

We know, in fact, that the two heats of the heating are connected by the expression:

$$q = Q + A\int_{v_0}^{v_1} p_0\,dv, \text{ or } q - Q = A\int_{v_0}^{v_1} p_0 dv.$$

The values of the heating of one molecule of gas and the knowledge of the numbers N of molecules disengaged by the combustion, enables us to translate into figures the equation (2). Further, this equation will give directly the heat of combustion sought for in degrees centigrade; if we put

$$T_0 = 273; \quad T_1 = 0 + 273,$$

it becomes then

$$q = \Sigma N\left[\frac{a\theta}{1{,}000} + \beta\frac{(\theta + 273)^2 - 273^2}{1{,}000^2}\right]$$

in large calories, and we can solve it algebraically. But when we have to determine a great number of temperatures, as in this investigation, it is preferable to employ the following artifice:

Briefly, it consists in finding a positive value of θ, so that the number of calories expressed by Σ equals the number of calories q of the calorific power.

Arrange then the table of the heating of one gaseous molecule from 0° to 100, 200° C., etc., from 100 to 100° for example. Extend the table to the perfect gases, to superheated steam, and to carbonic acid. This operation is easy by means of the

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Taking account of the equation $pv = RT$ and the definitions of Q and of q, there exists between the two specific heats at constant pressure and constant volume.

$$(Cp - Cv)\,(T_1 - T_0) = Ap_0\,(v_1 - v_0) = \frac{Ap_0v_0}{273}\,(T_1 - T_0),$$

or between the molecular heats:

$$Cp - Cv = \frac{Ap_0v^0}{273}.$$

Taking as units the kilogram and the decimeter:

$$A = \frac{1}{E} = \frac{1}{4{,}260}, \quad p = 103.33, \quad v_0 = 22{,}300, \quad Cp - Cv = 1.98,$$

about two calories.

Perfect gases, $q = 6.5\,\frac{T_1 - T_0}{1{,}000} + 0.6\,\frac{T_1^2 - T_0^2}{1{,}000^2}$

Superheated steam, $q = 6.5\,\frac{T_1 - T_0}{1{,}000} + 2.9\,\frac{T_1^2 - T_0^2}{1{,}000^2}$

Carbonic acid, $q = 6.5\,\frac{T_1 - T_0}{1{,}000} + 3.7\,\frac{T_1^2 - T_0^2}{1{,}000^2}$

formulas we have given. Further, the calculation can be very rapidly made; the squares

$$[T + 100]^2,\ [T + 200°]^2,\ [T + 300°],$$

etc., are immediately deduced, one from the other.

The numbers N, on the other part, being known, it is very evident that the table in question enables the value of the total heating of the residue of combustion (azote, carbonic acid, superheated steam), corresponding to 100, 200° C., etc.

Also, as we have always an idea approximately of the temperature sought for, it suffices to calculate the value of Σ for temperatures near θ, in order to determine the quantities of heat q_1 and q_2, such as

$$q_1 < q < q_2$$
$$\theta_1 < \theta < \theta_1 + 100.$$

The temperature sought for is comprised between θ and $\theta_1 + 100$, it differs from θ_1 by a quantity E, $\theta = \theta_1 + E$. Now E is easily calculated with sufficient precision.

In fact, betwen θ and θ_1, + 100 there can be admitted that the increase of the temperature is proportional to the increase in the quantity of heat, and that, in this interval, 1 calorie corresponds to an augmentation of temperature:

$$\frac{100}{q_2 - q_1}.$$

E, which corresponds to an increase of heat $q - q_1$, is, under these conditions, represented by

$$(q - q_1)\frac{100}{q_2 - q_1},$$

and the approximate value of θ is

$$\theta = \theta_1 + (q - q_1)\frac{100}{q_2 - q_1}.$$

The solution is not rigorous, but a greater exactness is useless.

A further reproach to this method of determination is that it does not take into account the dissociation of the superheated steam and of the carbonic acid. Regarding

this objection, however, there may be stated that the dissociation of the superheated steam between 0° and 2,500° has a negligible influence. The effect of the dissociation of the carbonic acid is, also, practically but little sensible up to 2,000°, a temperature near which are grouped the most part of the numbers that I give later on. Two of these numbers are near 2,500°, and it is proper to say that the coefficients of dissociation of the carbonic acid seem to have a real importance in the neighborhood of that temperature.*

II.—TABLE OF THE HEATINGS IN CALORIES.

The table of the heatings of the molecular volume of the gases is a useful document for many purposes. I think it

Heatings in calories of the molecular volume of gases from 0 degree centigrade to 3,000 degrees centigrade (under constant pressure.)

The molecular volume corresponds to 22.32 liters. The molecules are represented by the symbols Az^2, O^2, H^2, CO, CO^2.

Temperatures, degrees centigrade.	Azote, oxygen, oxide of carbon, hydrogen.	Superheated steam.	Carbonic acid.	Temperatures, degrees centigrade.	Azote, oxygen, oxide of carbon, hydrogen.	Superheated steam.	Carbonic acid.
0	0.00	0.00	0.00	1,600	12.46	20.35	23.09
100	0.68	0.83	0.87	1,700	13.34	22.13	25.18
200	1.39	1.73	1.85	1,800	14.23	23.93	27.31
300	2.10	2.67	2.87	1,900	15.14	25.83	29.55
400	2.82	3.69	3.99	2,000	16.05	27.76	31.83
500	3.56	4.76	5.17	2,100	16.98	29.74	34.18
600	4.31	5.89	6.44	2,200	17.92	31.81	36.64
700	5.07	7.07	7.77	2,300	18.87	33.91	39.14
800	5.85	8.30	9.16	2,400	19.84	36.10	41.75
900	6.63	9.62	10.66	2,500	20.81	38.32	44.40
1,000	7.43	10.98	12.12	2,600	21.80	40.62	47.16
1,100	8.24	12.40	13.85	2,700	22 80	42.95	49.96
1,200	9.05	13.87	15.55	2,800	23.82	45.37	52.87
1,300	9.89	15.41	17.33	2,900	24.84	47.82	55.81
1,400	10.73	17.00	19.18	3,000	25.88	50.35	58.86
1,500	11.59	18.65	21.11				

*The coefficient of dissociation is the proportion of the gas dissociated to that which would exist if the decomposition were complete. For carbonic acid, and at 2,500°, these coefficients are approximately 0.2 (pressure of 1 atmosphere), 0.4 (pressure of $\frac{1}{10}$ of an atmosphere), etc.

See Experimental Researches on Chemical Equilibriums, by Le Chatelier, "Annales des Mines," 1888.

beneficial to publish the one which I have used; its figures have been carefully collated, and it is more extensive and nearer complete than the most part of those heretofore published.

III.—EXAMPLES OF DETERMINATION OF HEATS OF COMBUSTION.

Our hypotheses suppose that the combustibles are burned under the normal atmospheric pressure and with the aid of air, at the temperature of 0° C.

One hundred volumes of air decompose into about 80 volumes of azote and 20 volumes of oxygen. Consequently, if a molecular volume of oxygen, O^2, is necessary for the burning of a certain combustible, the gas resulting from the operation will comprise, specially, 4 molecular volumes of azote or $4Az^2$.

In order to commence, take the simplest case, that of the combustion of hydrogen.

(1) *Temperature of Combustion of Hydrogen.*—According to the above conventions, the combustion of hydrogen in cold air will be expressed as follows:

$$H^2 + \frac{O^2}{2} + 2Az^2 = 2Az^2 + H^2O, \text{ gas}, + 58.2 \text{ calories.}$$

The residue of the combustion of 1 molecule (or 2 grams) of hydrogen comprises, then, 2 molecules of azote and 1 molecule of vaporized water. The heat disengaged by the reaction is 58.2 calories $= q$.

By referring this to the table of molecular heatings, there are easily obtained the values of Σ' at about the anticipated temperature θ of combustion.

Heating at (degrees),	1,800	1,900	2,000
Of 2 molecules of azote,	$2 \times 14.23 = 28.46$	$2 \times 15.14 = 30.28$	$2 \times 16.05 = 32.10$
Of 1 molecule of steam,	$= 23.93$	25.83	27.76
	52.39	$q_1 = 56.11$	$q_2 = 59.86$

The temperature sought for is, then, between 1,900° and 2,000° C., corresponding respectively to the heatings in calories 56.11 and 59.86. Finally, we have, by applying the

formula established above and taking account that $q=58.2$ calories,

$$\theta = 1{,}900 + (q - q_1)\frac{100}{q_2 - q_1} = 1{,}900 + 2.09\frac{100}{3.75} = 1{,}956^\circ \text{ C.},$$

or, in round numbers, 1,960° C.

We should find for the oxide of carbon, very approximately, 2,100° C., the flame of oxide of carbon is, then, hotter than the flame of hydrogen.

(2) *Temperature of the Combustion of a Peat.*—Analysis of 100 grams of the specimen remaining after abstraction of ash and hygroscopic water:

Carbon	57.21
Hydrogen	5.69
Oxygen and azote	37.10
	100.00

Calorific power, water vaporized, 5,590 calories, corresponding for 100 grams to 559 calories.

The gases resulting from the combustion of 100 grams, in air, of this peat comprise carbonic acid, water vaporized, and azote, of which the above analysis permits the determination of the number N of molecular volumes.

The proportion of carbonic acid is deduced from its tenor in carbon, as we know that 12 grams of carbon give, in burning, a molecular volume of carbonic acid, there will be in the gases of combustion

$$\frac{57.21}{12} = 4.76$$

volumes of carbonic acid.

Similarly, since 2 grams of hydrogen produce a molecule of water, the proportion of water vaporized will be

$$\frac{5.69}{2} = 2.84$$

volumes.

To calculate the azote, it is necessary to proportion it to the quantity of oxygen strictly utilized for the combustion.

	Volumes of oxygen.
Now, with 4.76 volumes of carbonic acid, there are combined.....	4.76
And, with 2.84 volumes of water vaporized, there are combined.....	1.42
Total.....	6.18

Deducting from this sum the corresponding volume of oxygen pre-existing in the peat

$$\left(\frac{37.10}{32} =\right) 1.16,$$

and neglecting the small quantity of azote which accompanies this body in the peat, it becomes $6.18 - 1.16 = 5.02$, or about 5 volumes. Five molecular volumes of oxygen are then necessary to burn the peat in question; these 5 volumes correspond to 25 volumes of air, containing 20 volumes of azote which pass unchanged into the gases of combustion.

To sum up: The residue of combustion will have the following composition in molecular volumes:

	Volumes.
Azote.....	20.00
Vaporized water.....	2.84
Carbonic acid.....	4.76
	27.60

Possessing thus the numbers N of gaseous molecules, we determine in the following manner the values of Σ in the neigborhood of the temperature of combustion sought for:

Heatings at (degrees)—	1,900.	2,000.	2,100.	2,500.	2,600.
Of 20.00 volumes of azote..	302.80	321.00	339.60	416.20	436.00
Of 2.84 volumes of steam...	73.35	78.83	84.46	108.80	115.40
Of 4.76 volumes of carbonic acid.....	140.65	151.51	162.69	211.34	224.50
	516.80	$q_1 = 551.34$	$q_2 = 586.75$	736.34	775.90

As on the other part, $q = 559$, the temperature of combustion sought for is visibly placed between 2,000° and 2,100°, we have, finally, with very close approximation,

$$\theta = 2{,}000 + (559 - 551)\frac{100}{587 - 551} = 2{,}020° \text{ C.}$$

The values above of the heatings at 2,500° and at 2,600° enable an idea to be had of the influence that would be exerted on the value of the flame by the previous heating of the air to 1,000° for example. In this hypothesis, the quantity q would be formed not only of the number of calories due to the heat of the combustion, but also of the number of calories carried by 25 volumes of air heated to 1,000°, $q = 559 + 25 \times 7.43$; according to the table of the heatings $q = 559 + 186 = 745$ calories. Completing the calculation, we find $\theta = 2{,}520°$. The previous heating of the air to 1,000° has raised the temperature of combustion about 500° C.

(3) *Temperature of Combustion of a Coal.*—Take, for example, the coal of Treuil (St. Etienne).

The elementary analysis of this coal has given abstraction made of ash and hygroscopic water:*

Supposing the water resulting from the combustion to be vaporized, the calorific power is 8,580 calories.

The method of calculation employed for the peat is here applicable without modification. We find, notably in the gaseous residue, the following composition in molecular volumes:

Azote	34.00
Superheated steam	2.50
Carbonic acid	7.40
Total	43.90
corresponding to 42.50 volumes of air used in the combustion.	
Value of the flame (cold air), degrees	2,010
Value of the flame (air at 1,000 degrees), degrees	2,570

I add that the reader will find other examples, notably in what concerns gaseous combustibles, in Mr. Damour's book, already cited.

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* Combustibles, such as coal and peat, always contain a certain quantity of hygroscopic water which passes entirely into the gases of combustion. It is very easy to take account of this water in the calculation of the temperature.

Carbon	89.23
Hydrogen	5.03
Oxygen and azote	5.74

IV.—RESULTS OF CALCULATIONS.

I have applied the preceding method to the calculation of the value of the flame of a certain number of combustibles. In order to render more easy the comparison of the results, I have supposed these combustibles to be deprived of their ash and hygroscopic water.

I have taken the elementary analyses and the calorific powers from pages 54, 59 and 80 of my work, entitled "Etudes sur les Combustibles." The following table gives the figures obtained.

The value of the flame varies very little from one combustible to another. The mean of sixteen enumerated is 1,986°, and it seems that the round number 2,000° can be adopted as representing the theoretical temperature of combustion of any combustible whatever of vegetable origin. As absolutely rigorous constants could not be obtained for the process I have described, a more precise number is useless.

Designation of the combustibles.	Calorific power.		Value of the flame.
	Water condensed.	Water vaporized.	
	Calories.	*Calories.*	*Degrees, C.*
Oak from Lorraine	4,690	4,370	1,865
Peat from Bohemia	5,900	5,590	2,020
Lignite from Trifail	6,650	6,370	1,960
Flaming coal from Blanzy (Ste. Marie)	8,350	8,060	1,990
Flaming coal from Decazeville (Tramont)	7,840	7,530	1,960
Oxidized coal from Commentry	6,380	6,200	1,960
Gas coal from Commentry	8,410	8,110	1,950
Gas coal from Béthune	8,670	8,380	1,990
Gas coal from Lens	8,740	8,450	2,010
Fat coal from St. Etienne (Treuil)	8,860	8,580	2,010
Marechale coal from Roche-la-Moleire	8,860	8,600	2,030
Semi-fat coal from d'Anzin (St. Marc)	8,660	8,430	1,980
Anthracite coal from Commentry	8,460	8,290	2,030
Anthracite coal from Kebao	8,530	8,370	2,020
Anthracite coal from Creusot	8,690	8,480	2,010
Anthracite coal from Pennsylvania	8,266	8,140	2,000

The value which, in the above table, is the farthest from the mean of 2,000°, is that of oak—1,865°.

The temperatures which are the farthest removed in excess from the mean 2,000° are, in general, those of the anthracite

coal. The popular opinion has always been that anthracite has a hotter flame than coal. The relative value of the flames sustains this opinion.

To the results in the above table can be added those which I have found for different liquid combustibles, as follows:

Ethylic alcohol and methylic alcohol, degrees..... 1,700
Amylic alcohol, degrees..... 1,850
Crude American petroleum .....2,000 degrees, (calorific power, 10,400.)
Essence of American petroleum.....1,910 degrees, (calorific power, 10,270.)
Refined American petroleum oil.....1,660 degrees, (calorific power, 10,280.)

Finally, below will be found the temperatures of combustion of several gases under constant pressure:

Hydrogen, degrees 1,960
Oxide of carbon, degrees..... 2,100
Methane, degrees..... 1,850
Acetylene, degrees 2,350
Illuminating gas, mean*, degrees..... 1,950
Water gas, industrial, degrees..... 2,000

An examination of the numbers grouped in this work shows the apparently paradoxical appearance that of two combustibles, the one having the greatest calorific power is not necessarily the one having the hottest flame.

The flame of the essence of petroleum is relatively cold. The flame of the sample of peat appears hotter than that of the coal of the Treuil, the calorific power of which is excellent. The flame of the oxidized coal from Commentry is a little better than that of the same coal not deteriorated.

Moreover, everything occurs in all cases, and, notably, for the combustibles of vegetable origin, as if the elements of the temperature of combustion, calorific power, elementary composition, volume and composition of the gases of combustion, compensated each other so as to give to all the combustibles flames of very nearly the same thermometric degrees of temperature.

This, however, does not lessen the advantages due to the high calorific power of good coals, and which is the true

*According to Mr. A. Sainte-Claire Deville (Société Technique de l'Industrie du Gaz," 1903.)

measure of their value, since the flame utilizable of a coal is a great deal more voluminous than that of a peat, a very much oxidized matter, the combustion of which requires but very little air.

We have seen, for example, that the combustion of 100 grams of Bohemian peat corresponds to the disengagement of 28 volumes of gases of combustion at 2,000°, while the gaseous residue of the coal from the Treuil attains 44 units* at the same temperature.

* The use of hot air for improving the flame, from the thermometric point of view, is certainly more efficacious in the furnaces in which coal is burned than in those in which an inferior combustible is burned.

—Reprinted from "Journal of the Franklin Institute."

TEST OF THE MACHINERY OF THE *JAMES C. WALLACE* UNDER SUPERHEATED AND SATURATED STEAM.

By Commander W. W. White, U. S. N. (Retired), Member.

The economical advantages in using superheated steam have been recognized for many years, but not until recently has steam engine and steam turbine construction advanced to the point where, from a practical standpoint, the ill effects met with in past years have been overcome. Formerly, the impossibility of obtaining satisfactory metallic and other packing for rods and joints subjected to high temperatures, with the numerous attendant evils, as well as a proper cylinder lubricant, served to delay its general use. Of late years, however, such difficulties have largely disappeared, and in modern practice a temperature of 500 degrees Fahrenheit is readily and successfully handled in steam engines. In foreign countries, notably in Germany, a much higher degree of superheat is being used, but the limit, thus far, in the United States appears to be as stated. This, no doubt, results from the additional expense involved in up-keep incidental to high temperatures.

For the purpose of securing comparative data with superheated and saturated steam, used in the same engine, and by courtesy of the Acme Steamship Company on the Great Lakes, a Board was appointed, under instructions from the Bureau of Steam Engineering, to carry out tests of machinery on the steamer *James C. Wallace.* This vessel, which is one of the largest freighters on the Lakes, has lately been put into service. She is equipped with two Babcock & Wilcox marine water-tubular boilers with superheaters, and the arrangement is such that the latter may be dispensed with by a few simple changes, and saturated steam used.

The Board was composed of Commander W. W. White, U. S. N. (Retired); Lieutenant Commander B. C. Bryan, U. S. N., and Lieutenant H. T. Winston, U. S. N.

A description of this vessel appeared in the JOURNAL, Vol. XVII, page 579. It will be only necessary, therefore, to repeat here the principal dimensions.

HULL.

The hull is of mild steel and was built by the American Shipbuilding Company, of Cleveland, Ohio. A cargo hold, hopper shaped, with girder arches, and without bulkheads, divisions or stanchions, constitutes an important departure and improvement in design. This hopper extends 408 feet in length, and measures 47 feet at the top and 29 feet at the bottom. The adoption of this construction makes it possible, by the use of automatic clams, to unload cargo without hand shoveling.

Length over all, feet	552
Beam, feet	56
Depth, feet	31
Number of cargo hatches	32
Capacity of water bottoms, tons	8,000
Coal bunker capacity, about, tons	350

MACHINERY.

The propelling machinery, as is usual in Lake vessels, is placed in the extreme after part of the vessel. The boilers are immediately forward of the engine, with an athwartship coal bunker forward of the boiler-room bulkhead.

ENGINE.

The main engine is of the quadruple-expansion, vertical, direct-acting, inverted, jet-condensing type. One piston valve, actuated by a Joy valve gear, is fitted on the side for each of the high, first and second intermediate-pressure cylinders. The low-pressure valve is a double-ported slide, placed on the after side of the cylinder and operated by eccentrics and a double-bar Stephenson link. All cylinders are unjacketed.

Number of cylinders	4
Diameter of H.P., inches	18½
1st I.P., inches	28½
2d I.P., inches	43½
L.P., inches	66
Stroke (common), inches	42
Diameter of piston rods (one for each cylinder), inches	5⅝
Order of cylinders from forward: (1) H.P.; (2) 1st I.P.; (3) 2d I.P.; (4) L.P.	
Designed I.H.P	2,000
piston speed, feet	560

AUXILIARY MACHINERY.

The following shows the important dimensions of the auxiliary machinery in use during the tests, all of which is independent of the main engine. In addition, in Test II, two fireroom blowers (double simple engine 6 inches × 5 inches), and two small duplex pumps (2 inches × 1¼ inches × 1⅜ inches stroke), which furnished water circulation through the blower bearings, were in operation.

Number.	Purpose.	Make.	Type.	Diameter of cylinders.			Stroke.
				H.P.	L.P.	Water.	
1	Main air pump.	Blake.	Vertical compound.	12	20	24	18
1	Main feed pump.	Blake.	Horizontal, compound, duplex plunger.	6	10	5	10
1	Water service pump.	Blake.	Horizontal duplex.	...	4½	3⅜	4
1	Bilge pump.	Blake.	Horizontal duplex.	...	7½	8½	6
1	Stoker engine.	...	Vertical, one-cyl., simple.	...	6½	...	6

A feed-water heater is fitted in the engine room, into which all auxiliaries exhaust. The heater is cylindrical, with 2-inch tubes, through which the feed water is forced in its passage to the boilers. No attempt was made to determine the power of any of the auxiliaries in use.

BOILERS.

Steam is supplied by two Babcock & Wilcox boilers of the latest type, designed for a pressure of 250 pounds, with super-

heaters and stokers. Plate I shows a sectional, and Plate II a front view of one of the boilers.

The superheater, it will be observed, consists of an upper and lower cross box having a number of bent tubes expanded into them. Steam, after being generated, is led from the drum by an outside pipe to the forward end of the superheater. After passing through the latter, it is directed through another outside pipe, at the after end, to the main steam pipe. One partition is placed in each cross box, thus causing the steam to flow three times from one box to the other in its passage from inlet to outlet.

Suitable valves are installed in the piping, so that the generated steam may be sent either through the superheater, or, if desired, directly to the main steam line.

Length of boiler proper, feet	11
including projection of stoker, feet	14
Width of boiler, feet and inches	12– 2
Height of boiler to top of drum, feet	15
superheater, feet and inches	15– 8
Diameter of drum, inches	42
Length over heads, feet and inches	13– 5
Thickness of shell plates, inch	$\frac{1}{2}^{\frac{9}{16}}$
Diameter of boiler tubes, inches	4
Thickness of boiler tubes	6 B. W. G.
Length of tubes between headers, feet	10
Number of sections wide, each boiler	18
including side sections	20
Diameter of superheater tubes, inches	2
Thickness of superheater tubes	10 B. W. G.
Square feet of grate surface, each boiler	74
heating surface, each boiler	2,900
superheating surface, each boiler	414
Ratio of G. S. to H. S.	1 to 39.19
including superheater	1 to 44.78

STOKERS.

Each boiler is provided with two chain stokers of the Crowe pattern, all driven by a single-cylinder engine $6\frac{1}{2}$ inches $\times$ 6 inches (installed in the engine room and run at constant speed) through shafting, sprocket wheels, chain, eccentrics, &c. The speed of each stoker grate is independently controlled by an

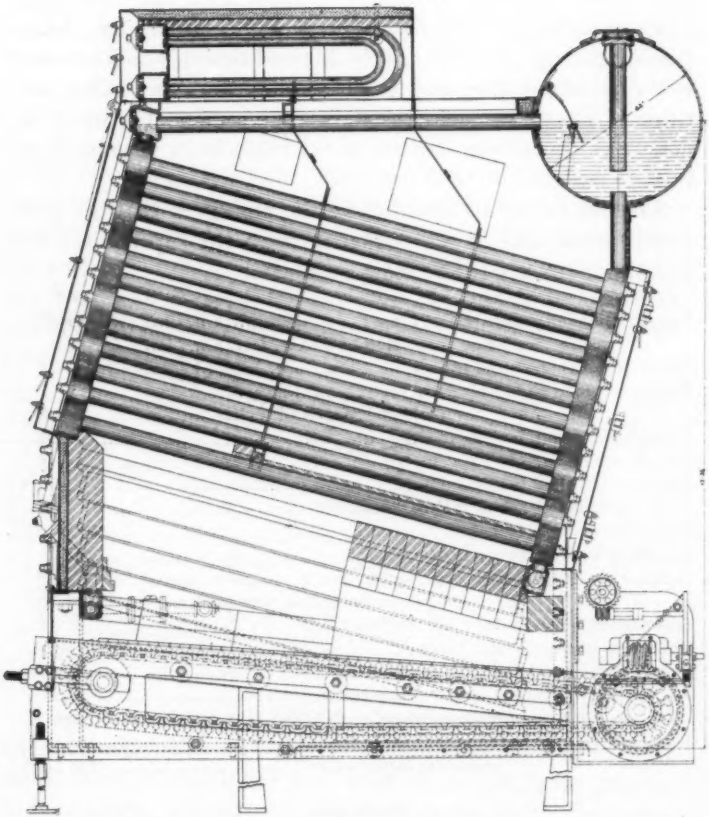

Plate I.—BABCOCK & WILCOX BOILERS, S. S. "JAMES C. WALLACE."

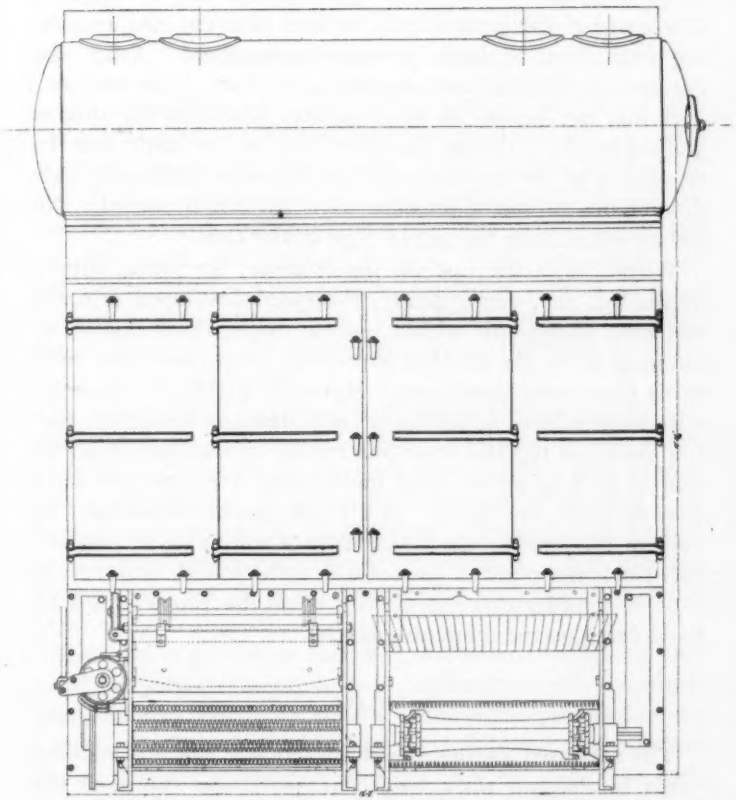

Plate II.—BABCOCK & WILCOX BOILERS, "S. S. "JAMES C. WALLACE."

adjustable ratchet device, and its rate of travel is regulated, as frequently as circumstances require, in order to maintain steam at the proper pressure. Coal is fed into hoppers, at the front of the boilers, the thickness of the fire being governed by an adjustable plate extending across the width of the grate. The speed of the grate should be such that the coal, in passing from front to back, is entirely consumed. Ashes and clinkers are dumped and removed at the back of the boilers.

It may not be out of place to note here that the stokers worked perfectly during the entire trip, and so easily was the regulation of the coal supplied the furnaces controlled, that fluctuations in steam pressure were but slight indeed. No smoke issued from the smoke pipe at any time.

Where, as is the case on the *Wallace*, the steam supply demanded, and, consequently, the speed developed, is fixed, and only changeable within narrow limits, there can be no question as to the decided superiority of stokers over hand firing from every standpoint. Moreover, practical experience with stokers fitted to boilers of Lake steamers during the past few years has resulted in improving the mechanical construction to such an extent that breakdowns are rare, and when they do occur, are usually readily and quickly remedied. In view of this success, the advisability of equipping at least one-half the boiler plant of men-of-war with stokers is worthy of serious consideration; flexibility could be provided for by hand firing on the remaining boilers.

COAL.

The coal used during the entire trip was western Pennsylvania bituminous slack, loaded at Ashtabula. On each test an average sample was secured and sealed in a jar for determination, at a convenient time, of moisture, heat value, &c.

Three samples have been examined, and the heat value in British thermal units (by Mahler bomb calorimeter) per pound of dry coal, and proximate analyses are recorded below:

	B.T.U.	Percentages of dry coal.		
		Volatile matter.	Fixed carbon.	Ash.
Sample No. 1,	13,466	29.23	59.53	11.23
Sample No. 2,	13,499	28.37	57.51	14.42
Sample No. 3,	13,593	28.60	60.63	10.77
Average,	13,519	28.73	59.22	12.14

DESCRIPTION OF TESTS.

For comparison, two tests (tables I and II) were made with the superheater in operation, and two (tables III and IV) with saturated steam, the superheater being entirely blocked off and cut out. During all tests, the coal was carefully weighed, the main engine indicated, and the important engine and fire-room data recorded half hourly. The water evaporated was not weighed, as the facilities for doing so were lacking. This is to be regretted, since valuable information regarding steam economy of the main engine under the different conditions was not obtainable.

A blower, having a central inlet and discharging at its periphery, is fitted (see plates III and IV) at the back and over each boiler, to handle the products of combustion after passing through the boiler. In the ordinary steaming of the ship these blowers are used, as the draft is thus under complete control, and entirely independent of weather conditions. These blowers, however, are not an absolute necessity, and may or may not be used.

Plate III is an outline sketch of the boiler, in which is indicated the path of the products of combustion with the superheater in operation. As shown, a damper is provided in the uptake, which is kept closed or open, depending on whether the blowers are running or stopped. Of the two tests with superheated steam, the first was conducted with the blowers stopped, and the second with blowers (plate III) running. Both of these tests were made on the up trip of the vessel (Ashtabula, Ohio, to Duluth, Minn.) with no cargo, but with water ballast.

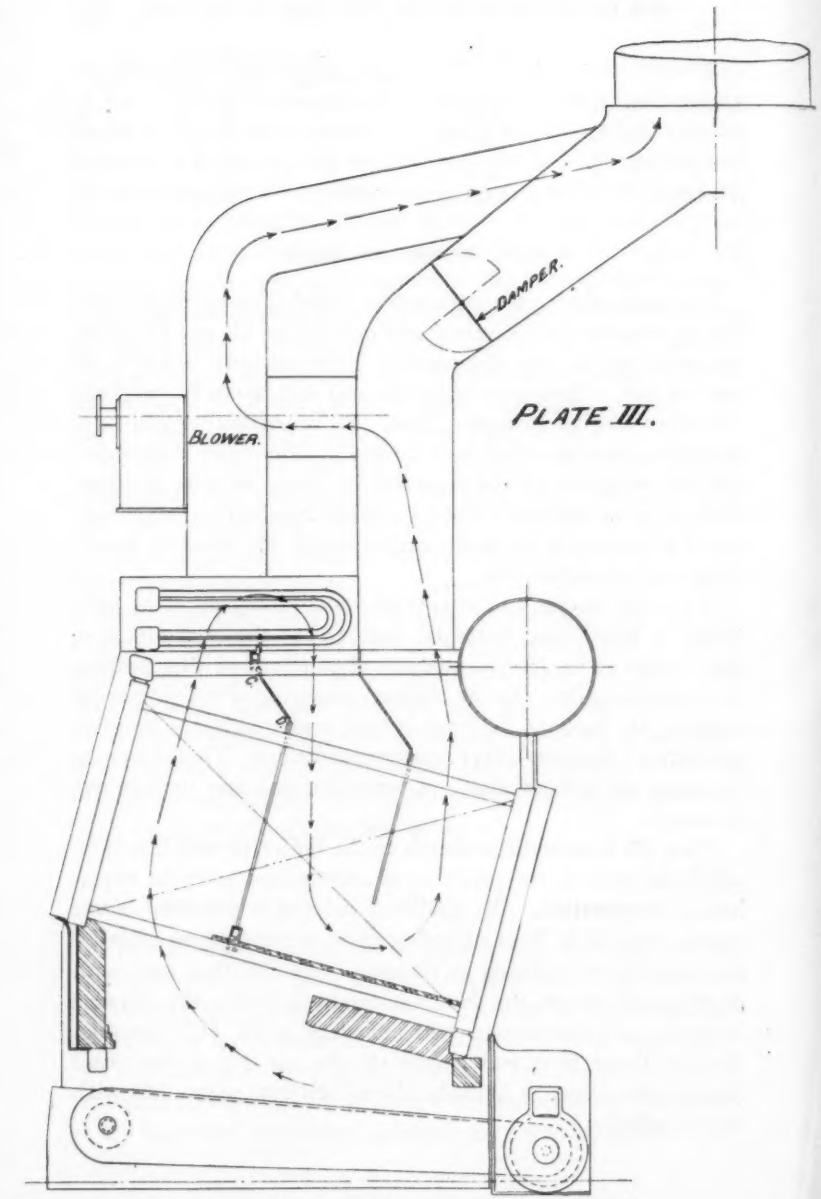

PLATE III.

INDICATING PATH OF GASES OF COMBUSTION WITH SUPERHEATER IN OPERATION AND USING INDUCED DRAFT.

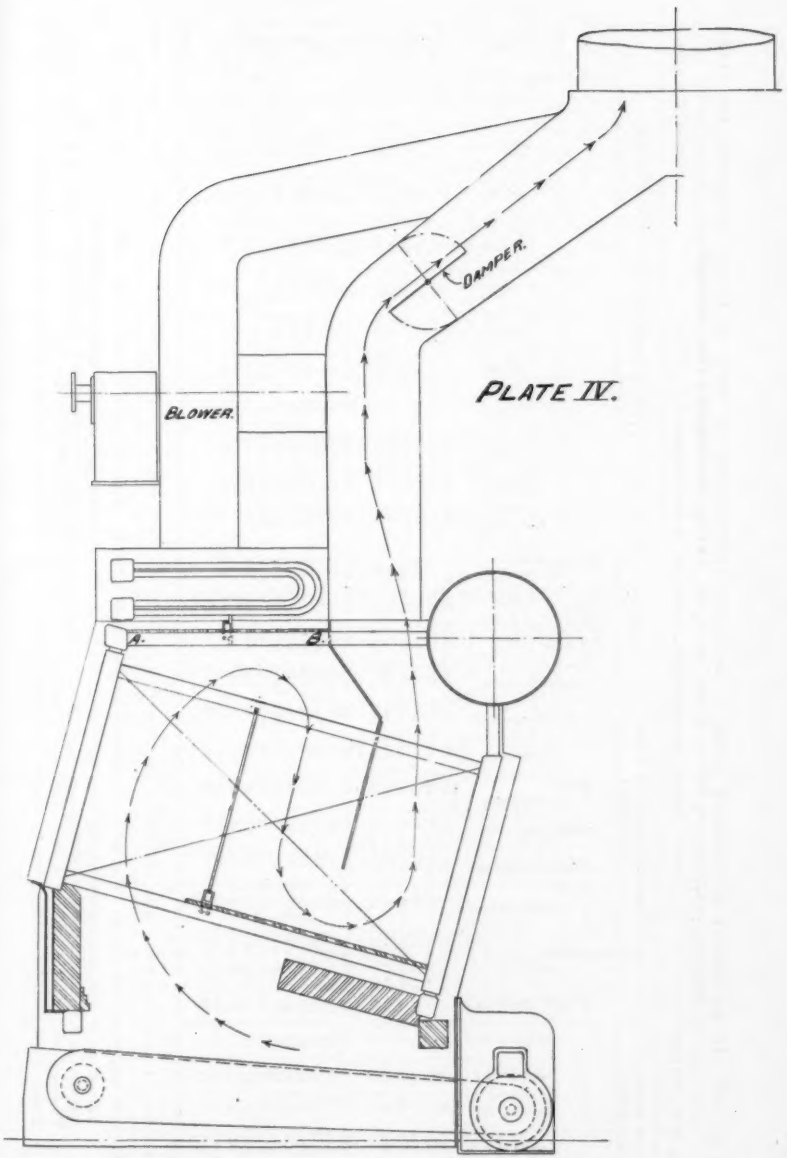

PLATE IV.

INDICATING PATH OF GASES OF COMBUSTION WITH SUPERHEATER CUT OUT AND BLOWERS STOPPED.

Table I.—TEST OF MACHINERY OF STEAMER *JAMES C. WALLACE* (CONTROLLED BY ACME STEAMSHIP COMPANY), ON LAKE HURON, JULY 29, 1905, 9·30 A. M. TO 5·30 P. M., **USING SUPERHEATED STEAM.**

Duration of test, 8 hours. Draught, forward, 6 feet 4 inches; aft, 14 feet 8 inches. Speed, 12.6 miles.

Time.	Steam pressures per gauge.					Vacuum (inches).	Reading of engine counter.	Revolutions or double strokes per minute.						Temperatures, degrees, F.						M. E. Pressures.				I.H.P.					Coal.
	At boiler.	At engine.	1st receiver.	2d receiver.	3d receiver.			Main engine.	Main air pump.	Main feed pump.	Water-service pump.	Bilge pump.	Stroke, engine.	Steam at engine.	Steam leaving superheater. Port boiler.	Steam leaving superheater. Str'b'd boiler.	Of gases at base of stack.	Injection.	Hotwell.	Feed.	H.P.	1st I.P.	2d I.P.	L.P.	H.P.	1st I.P.	2d I.P.	L.P.	Total.	
A. M.																													
9 30	237	233	93	34	10	24.5	154,819	...	25	18	48	16	134	478	484	483	...	66	104	194	...	...	...	...	...	...	...	...	...	
10 00	237	235	93	34	9.7	24.7	157,172	78.4	26	18	45	19	136	480	484	483	...	66	104	194	89	36	16.1	6.8	381	375.1	394.8	385.6	1,536.5	
10 30	237	234	94	34.5	10	24.6	159,531	78.6	26	16	48	18	140	474	484	483	487	66	104	192	91	36.2	16.2	6.9	390.6	378.1	398.2	392.3	1,559.2	
11 00	237	231	95	34	9.8	24.7	161,898	78.9	28	16	46	18	144	482	484	483	490	66	104	200	88.5	37	15.9	6.8	381.3	388	392.4	388	1,549.7	
11 30	237	233	93	34	9.7	24.6	164,254	78.5	28	15	48	19	142	484	486	486	485	66	104	198	91	36.8	15.6	6.7	390.2	383 9	383	380.4	1,537.5	
12 00	237	234	93	34	9.8	24.6	166,612	78.6	28	18	48	18	142	480	486	486	483	66	106	196	90	36.8	16.1	6.8	386.3	384.4	395.8	386.6	1,553.1	
P. M.																													
12 30	229	228	94	34	10	24.6	168,959	78 2	27	18	48	18	138	478	490	488	486	64	104	194	88.5	37	16	6.6	378	384 5	391.3	373.3	1,527.1	
1 00	237	230	94	34	10	24.4	171,321	78.7	27	18	48	18	140	479	489	488	480	64	104	209	88	36.5	15.9	6.5	378.2	381.7	391.4	370	1,521.3	
1 30	237	233	94	34	10	24.6	173,690	79	28	15	48	17	144	484	489	488	465	64	102	194	91.5	37	16	6.9	394.8	388.5	395.3	394.3	1,572.9	
2 00	237	231	94	34	10	24.5	176,066	79.2	27	17	48	17	144	484	489	488	481	64	104	194	89	36	16	6.8	385	378.9	396.4	389.5	1,549.8	
2 30	237	234	94	33	10	24.5	178,431	78.8	25	16	46	17	142	483	489	488	493	63	104	195	89	35	16.1	6.75	383	366.5	396.8	384 7	1,531	
3 00	237	232	94	34	10	24.5	180,807	79 2	27	16	48	16	144	484	489	488	474	61	102	192	91	35.6	16.2	6.8	393.6	374.7	401.3	389.5	1,559.1	
3 30	234	234	95	34	10	24.5	183,170	78.8	27	16	48	16	144	487	487	488	...	62	102	193	91.5	36	16 2	6.8	393.8	377	399.3	387.5	1,557.6	
4 00	239	232	95	34	10	24.5	185,563	79.8	26	16	48	17	144	484	489	488	502	63	104	194	91	35 5	16	7	396.6	376.5	399.4	404	1,576.5	
4 30	242	240	96	35	10.2	24.5	187,956	79.8	26	16	48	16	140	486	490	489	...	63	105	196	93	36.5	16.1	7.3	405.3	387.1	401.8	421.4	1,615.6	
5 00	242	240	96	35	10.2	24.6	190,355	80	25	16	48	16	138	485	489	488	498.5	63	104	190	93.5	38	16.5	7.2	408.5	404	412.9	416 6	1,642	
5 30	242	239	95	34	10	24.6	192,749	79.8	26	16	46	16	138	486	489	488	...	62	104	192	...	...	...	...	...	...	...	...	...	
Total.	4,040	3,973	1,602	579.5	169.4	417.5	37,930	1,264 3	452	281	807	294	2,394	8,198	8,287	8,273	5,824.5	1,089	1,765	3,317	1,355 5	545.9	240.9	102.65	5,846.2	5,728 9	5,950.1	5,863.7	23,388.9	*
Av'ge	237.6	233.7	94 2	34.1	9.96	24.5	79 02	79.02	26.6	16.5	47 5	17.17	140.8	482.2	487 4	486.6	485.3	64 06	103.8	195 1	90.36	36.39	16.06	6.84	389.7	381.9	396.7	390.9	1,559.2	

* Total coal (as weighed) = 18,900 pounds. Average coal (as weighed), per hour = 2,362.5 pounds. Moisture in coal, per cent. = 3.38. Ashes in coal, per cent. = 19 87.

Average number of degrees superheat at engine..... 81.7° F.
Average I.H.P. during test (main engine only)..... 1,559.2
Pounds of coal per I.H.P. per hour, as fired..... 1.515
Pounds of coal per I.H.P. per hour, dry..... 1.464

Average draft (inches of water) at base of smoke pipe, = 0.22
Average temperature engine room, lower platform, = 79° F.
Average temperature fireroom, = 81° F.
Test made under natural draft—fireroom blowers stopped.

Table II.—TEST OF MACHINERY OF STEAMER *JAMES C. WALLACE* (CONTROLLED BY ACME STEAMSHIP COMPANY, ON LAKE SUPERIOR, JULY 30, 1905, 10 A. M. TO 4 P. M., **USING SUPERHEATED STEAM.**

Duration of test, 6 hours. Draught, forward, 6 feet 4 inches; aft, 14 feet 8 inches. Speed, 12.7 miles.

Time.	Steam pressures per gauge.					Vacuum (inches).	Reading of engine counter.	Revolutions or double strokes per minute.							Temperatures, degrees, F.							M. E. Pressures.				I.H.P.					Coal.*
	At boiler.	At engine.	1st receiver.	2d receiver.	3d receiver.			Main engine.	Main air pump.	Main feed pump.	Water-service pump.	Bilge pump.	Stoker engine.	Fireroom blowers.	Steam at engine.	Steam leaving superheater. Port boiler.	Steam leaving superheater. St'b'd boiler.	Of gases at base of stack.	Injection.	Hotwell.	Feed.	H.P.	1st I.P.	2d I.P.	L.P.	H.P.	1st I.P.	2d I.P.	L.P.	Total.	
A. M.																														
10'00	244	239	97	35	10.2	24.3	253,200	...	24	16	24	22	132	168	488	473	490.5	510	60	114	205	89	35.9	16.2	7	...	...	...	...	...	
10'30	244	238	95	34	9.8	24.8	255,582	79.4	25	17	46	20	132	144	492	496	493.5	.....	58	105	195	90.1	35.6	16.2	6.9	390.7	375.6	402 3	396.3	1,564.9	
11'00	240	244	96	35	10	24 7	257,969	79 6	26	18	48	33	132	85	494	496	493.5	509	58	104	196	91.5	36.1	16 45	6 9	397.8	381 9	409.6	397.4	1,585.7	
11'30	244	239	95	34	9.8	24.8	260,356	79.6	25	17	40	32	130	136	491	496	495.5	.....	58	102	197	91.5	35 8	16.4	6.9	397 8	378.7	408 3	397.3	1,582.1	
12'00	242	239	95	34	9.8	24.9	262,744	79 6	24	15	34	31	118	120	491	498	495.5	487	54	99	196	91.5	35 2	16.4	6.95	397 8	372.4	408 4	400.2	1,578.8	
P. M.																														
12'30	235	239	95	34	9 5	24.8	265,124	79 3	27	17	35	30	128	136	491	496	493.5	492	55	102	198	93	35.6	16.7	6.9	402.7	375.2	415 8	395.8	1,589.5	
1'00	240	244	95	34	9.5	24 9	267,493	79	27	16	38	32	132	140	490	494	491.5	.....	53	98	200	92	35.9	17.2	6.95	396.9	376.9	425	397.2	1,596	
1'30	242	237	94	34	9.5	25.1	269,878	79.5	27	15	38	30	134	126	488	494	493.5	493	50	95	190	89	35.6	16 7	6 95	386.4	376.2	415.3	399.7	1,577.6	
2'00	241	242	96	34	9.8	25.2	272,273	79.8	26	16	36	30	126	110	492	496	493.5	.....	46	92	193	91	35.6	16.5	7.1	396.6	377 6	411.8	409 8	1,595.8	
2'30	238	238	95	34	9.5	25.2	274,668	79.8	26	14	34	30	124	136	493	498	495.5	512	45	91	192	91	35 5	16.4	7.1	396.6	376.5	409 4	409.8	1,592.3	
3'00	242	245	96	35	10	24.8	277,061	79.8	25	15	35	30	125	150	492	500	496.5	.....	43	103	198	92	35 8	16.8	7 1	400 9	379.7	419.3	409.8	1,609.7	
3'30	242	243	96	35	9 8	25	279,460	80	26	13	36	30	126	136	485	498	495.5	490	44	103	200	91	36	16 6	7	397.6	382.8	415.4	405.1	1,600.9	
4'00	242	238	95	34	9 5	25	281,854	79 8	26	16	35	30	124	144	4 10	498	495.5	504	43	102	192	93	35.5	16 7	7	405.3	376.5	416 8	404.1	1,602.7	
Total,	3,136	3,125	1,240	446	126.7	323.5	28,654	955 2	334	205	479	379	1,663	1,731	6,381	6,453	6,423.5	3,997	667	1,310	2,552	1,185.6	464.1	218 25	90.75	4,766.1	4,530	4,957 4	4,822.5	19,076	
Av'ge	241.2	240.4	95.4	34.3	9 74	24.88	79 6	79 6	25.7	15 7	36 8	29.15	128	133 1	490.8	496 4	494.1	499 6	51.3	100.77	196 3	91.2	35.7	16.78	6 98	397.1	377.5	413.11	401.9	1,589.6	

* Total coal (as weighed) = 15,700 pounds. Average coal (as weighed) per hour = 2,616.7 pounds. Moisture in coal, per cent. = 2.89. Ashes in coal, per cent. = 15.32.

Average number of degrees superheat at engine..... 88° F.
Average I.H.P. during test (main engine only)..... 1,589.6
Pounds of coal per I.H.P. per hour, as fired..... 1.646
Pounds of coal per I.H.P. per hour, dry..... 1.598

Average draft (inches of water) at base of smoke pipe = 0 42
Average temperature engine room, lower platform, = 77° F.
Average temperature fireroom, = 76° F.
Test made with induced draft—fireroom blowers in operation.

Table III.—TEST OF MACHINERY OF STEAMER *JAMES C. WALLACE* (CONTROLLED BY ACME STEAMSHIP COMPANY), ON LAKE SUPERIOR, AUGUST 3, 1905, 8 A. M. TO 4 P. M., **USING SATURATED STEAM.**

Duration of test, 8 hours. Draught, forward, 19 feet 4 inches; aft, 19 feet 5 inches. Speed, 11.9 miles.

Time.	Steam pressures per gauge.					Vacuum (inches).	Reading of engine counter.	Revolutions or double strokes per minute.						Temperatures, degrees, F.				M. E. pressures.				I.H.P.					Coal.
	At boiler.	At engine.	First receiver.	Second receiver.	Third receiver.			Main engine.	Main air pump.	Main feed pump.	Water-service pump.	Bilge pump.	Stoker engine.	Of gases at base of stack.	Injection.	Hotwell.	Feed.	H.P.	First I.P.	Second I.P.	L.P.	H.P.	First I.P.	Second I.P.	L.P.	Total.	
A. M.																											
8:00	237	236	107	41	11.8	24.5	492,093	...	20	24	27	12	150	...	50	107	185	82.5	39.5	19.5	8	...	...	...	...	...	
8:30	237	238	108	41	12	24.5	494,485	79.7	19	23	34	14	144	554	50	105	185	83	40.5	19	8.05	361.3	429	473.7	464.1	1,728.1	
9:09	237	237	107	41	12	24.8	496,884	80	18	17	40	13	145	...	46	97	175	82.5	39	19	8.1	360.4	414.7	475.4	468.7	1,719.2	
9:30	235	234	108	41	12	24.5	499,280	79.9	19	20	36	13	145	548	46	104	184	81	41.5	19.1	8	353.5	440.7	477.4	462.4	1,734	
10:00	235	229	106	40	11.5	24.6	501,671	79.7	21	12	38	12	120	...	43	104	178	80	40.5	19.2	8	348.2	429	478.6	461.2	1,717	
10:30	239	235	108	41	12.1	24.6	504,064	79.7	20	18	38	12	144	549	43	106	187	82.5	41.2	19.2	8.1	359 1	436.4	478 6	467	1,741.1	
11:00	237	233	106	41	12	24 6	506,460	79.9	20	12	36	12	144	...	43	105	191	80.1	40	19	8.2	349.6	424 8	474.9	473.9	1,723.2	
11:30	237	234	101	37	11	24.6	508,882	80.7	24	14	36	11	142	584	42	106	186	88	41	19	7 5	387.8	439.7	479.6	437.8	1,744.9	
12:00	235	230	103	38	11	24.5	511,288	80.2	20	16	35	11	142	...	42	105	180	88	40.6	18	8	385.5	432.7	451.5	464.2	1,733.9	
P. M.																											
12:30	239	237	97	36	11.7	24.6	513,711	80.8	19	18	38	11	140	557	40	104	182	90	41.5	18	8.2	397.2	445.6	454.9	479.2	1,776.9	*
1:00	239	242	95	35	11.2	24.8	516,127	80 5	20	18	36	10	140	...	40	104	183	93	42.5	18	8.2	408.9	454.7	453.2	477.5	1,794.3	
1:30	241	233	95	35	11.5	24.5	518,554	80.9	21	20	38	11	144	578	41	105	180	86	40.5	17.5	7.9	380	435.5	442.8	462.3	1,720.6	
2:00	237	236	97	36	11.6	24 5	520,982	80.9	20	19	38	11	144	...	41	105	184	88.5	42	18	8.2	391	451.6	455.5	479.8	1,777.9	
2:30	232	226	91	33	10.7	24.6	523,409	80.9	20	20	42	10	142	534	42	104	179	88	40	17	7.9	388.8	430.1	430.2	462.3	1,711.4	
3:00	237	234	95	35	11.5	24.5	525,804	79.8	22	21	34	11	142	...	43	106	184	88	41.5	17.8	8.2	383.5	449.2	444.3	473.3	1,741.3	
3:30	242	240	98	37	12.1	24.4	528,224	80.7	21	21	36	12	136	553	44	109	190	88	41.5	18	8.4	387.9	445.1	454.4	490.4	1,777.8	
4:00	239	235	98	37	12.1	24.5	530,657	81.1	21	21	36	11	136	...	44	108	184	91	41.5	18.2	8.5	403	447.3	461.6	498.6	1,810.5	
Total,	4,035	3,989	1,720	645	197.8	417.6	38,564	1,285.4	345	314	618	197	2,400	4,457	740	1,784	3,117	1,460.1	694.8	313.5	137.45	6,045.7	6,997.1	7,386.6	7,522.7	27,952.1	
Av'ge	237.3	234.6	101.17	38	11.63	24.56	80.34	80.34	20.3	18.47	36 94	11.59	141.17	557.1	43.53	105	183.3	85.9	40.87	18.44	8.08	377.8	437.3	461.6	463.8	1,740.5	

* Total coal (as weighed) = 25,100 pounds. Average coal (as weighed) per hour = 3,137.5 pounds. Moisture in coal, per cent. = 3.07. Ashes in coal, per cent. = 17.8.

Average I.H.P. during test (main engine only).....1,740.5
Pounds of coal per I.H.P. per hour, as fired..... 1.803
Pounds of coal per I.H.P. per hour, dry..... 1.747
Ship loaded with 10,400 tons (2,240 pounds) of iron ore.

Average draft (inches of water) at base of smoke pipe = 0.24
Average temperature engine room, lower platform, = 70° F.
Average temperature fireroom, = 73° F.
Test made under natural draft—fireroom blowers stopped.

Table IV.—TEST OF MACHINERY OF STEAMER *JAMES C. WALLACE* (CONTROLLED BY ACME STEAMSHIP COMPANY), ON LAKE HURON, AUGUST 4, 1905, 8·15 A. M. TO 2·15 P. M., **USING SATURATED STEAM.**

Duration of test, 6 hours. Draught, forward, 19 feet 4 inches; aft, 19 feet 5 inches. Speed, 11.2 miles.

Time.	Steam pressures per gauge.					Vacuum (inches).	Reading of engine counter.	Revolutions or double strokes per minute.						Temperatures, degrees, F.				M. E. pressures.				I.H.P.					Coal.
	At boiler.	At engine.	1st receiver.	2d receiver.	3d receiver.			Main engine.	Main air pump.	Main feed pump.	Water-service pump.	Bilge pump.	Stoker engine.	Of gases at base of stack.	Injection.	Hotwell.	Feed.	H.P.	1st I.P.	Second I.P.	L.P.	H.P.	1st I.P.	2d I.P.	L.P.	Total.	
A. M.																											
8·15	239	240	99	37	10.5	24.4	585,601	...	18	15	54	34	146	...	63	106	194	81	37.5	19	7 25	...	...	...	...	...	
8·45	239	242	97	36	10.3	24.5	587,961	78.7	23	16	52	37	144	572	63	104	193	84	35·7	17 4	7.25	361	373.4	428.4	412.7	1,575.5	
9·15	238	240	89	34	10.2	24.6	590,307	78.2	18	15	52	35	124	...	63	104	192	81	35.4	16.9	7.2	345.9	367.8	413.4	407.3	1,534.4	
9·45	239	243	89	33	10.1	24.7	592,651	78.1	19	18	50	36	126	554	63	103	192	80	35.6	17 2	7.2	341.2	369.5	420.1	406 7	1,537.5	
10·15	240	239	89	33	10	24.7	594,996	78.2	20	20	51	36	130	...	63	104	191	80	35.8	17.1	7.1	341.6	372	418.3	401.6	1,533.5	
10·45	239	243	89	33	10	24.7	597,321	77.5	20	16	44	40	126	558	63	104	191	83.5	36	17.1	7.15	353 4	370.8	421.8	400.8	1,546.8	
11·15	239	238	89	33	10	24.4	599,596	75.8	20	15	36	42	104	...	63	104	190	81.5	35 9	17 1	7.15	337.4	361.6	405.5	392	1,496.5	
11·45	239	237	89	33	10.1	24.5	601,874	75.9	24	16	36	38	126	532	64	104	195	84	36.1	17	7 1	348.2	384.3	403.6	389.8	1,525.9	
P. M.																											
12·15	239	236	89	33	10.1	24.6	604,192	77.3	26	18	36	38	134	...	64	104	195	83.5	36.5	17	7 1	352.5	375	411	397	1,535.5	*
12·45	239	236	89	33	10.1	24 7	606,504	77.1	25	15	42	36	126	544	64	104	194	84	35.6	17	7.1	353.7	364.8	410	396	1,524.5	
1·15	237	235	89	33	10.1	24.6	608,785	76	22	16	40	38	126	...	64	104	192	85	38.5	17	7.2	352.8	388.9	404.1	395.8	1,541.6	
1 45	237	230	87	32	9.8	24.7	611,059	75.8	22	16	36	38	120	561	64	104	195	84	35 7	17	7	347.7	359.6	403 1	383.8	1,494.2	
2·15	239	233	89	33	10.2	24.6	613,324	75.5	22	15	38	38	120	...	63	104	192	84.5	36.5	17.4	7.25	348.4	366 2	410.9	395.9	1,521.4	
Total,	3,103	3,092	1,173	436	121.5	319.7	27,723	924.1	279	211	567	486	1,652	3,321	824	1,353	2,506	1,076	472.8	224.5	93 05	4,183.8	4,453.9	4,950.2	4,779.4	18,367.3	
Av'ge	238.7	237.8	90.2	33.5	10.12	24.6	77.01	77.01	21.46	16.23	43.61	37.4	127.1	553.5	63.4	104.1	192.7	82.77	36.37	17.27	7.16	348.6	371.1	412.5	398.3	1,530.5	

* Total coal (as weighed), = 16,200 pounds. Average coal (as weighed), per hour = 2,700 pounds. Moisture in coal, per cent, = 2.88. Ashes in coal, per cent. = 14.4.

Average I.H.P. during test (main engine only).....1,530.5
Pounds of coal per I.H.P. per hour, as fired..... 1.764
Pounds of coal per I.H.P. per hour, dry..... 1.713
Ship loaded with 10,400 tons (2,240 pounds) of iron ore.

Average draft (inches of water) at base of smoke pipe, = 0.22
Average temperature engine rooom, lower platform, = 84° F.
Average temperature fireroom, = 79° F.
Test made under natural draft—fireroom blowers stopped.

SUMMARY OF TESTS.

	Test I.	Test II.	Test III.	Test IV.
Duration, hours	8	6	8	6
Kind of steam	Super-heated.	Super-heated.	Satu-rated.	Satu-rated.
Steam pressures, gauge:				
At boilers	237.6	241.2	237.3	238.7
At engines	233.7	240.4	234.6	237.8
At first receiver	94.2	95.4	101.2	90.2
At second receiver	34.1	34.3	38	33.5
At third receiver	9.96	9.74	11.63	10.12
Vacuum, inches	24.5	24.9	24.6	24.6
Revolutions per minute	79	79.6	80.3	77
Cut off in cyls., per cent. of stroke:*				
H.P. cylinder	.574	.574	.574	.504
First I.P. cylinder	.659	.659	†.647	.698
Second I.P. cylinder	.598	.598	†.606	.64
L.P. cylinder	.605	.605	†.556	.58
Mean effective pressures:				
H.P. cylinder	90.36	91.2	85.9	82.77
First I.P. cylinder	36.39	35.7	40.87	36.37
Second I.P. cylinder	16.06	16.78	18.44	17.27
L.P. cylinder	6.84	6.98	8.08	7.16
Indicated horsepower:				
H.P. cylinder	389.7	397.1	377.8	348.6
First I.P. cylinder	381.9	377.5	437.3	371.1
Second I.P. cylinder	396.7	413.1	461.6	412.5
L.P. cylinder	390.9	401.9	463.8	398.3
Total	1,559.2	1,589.6	1,740.5	1,530.5
Coal:				
Moisture in per cent.	3.38	2.89	3.07	2.88
Per hr., as fired, pounds	2,362.5	2,616.7	3,137.5	2,700
Per hr. per I.H.P. (moist), lbs.	1.515	1.646	1.803	1.764
Per hr. per I.H.P. (dry), pounds.	1.464	1.598	1.747	1.713
Per hr. per sq. ft. grate (moist), lbs	15.96	17.68	21.2	18.24
Per cent. of ashes in coal	19.87	15.32	17.8	14.4
Kind of draft	Natural.	Induced.	Natural.	Natural.
Draft (ins. water), base of smokpipe	.22	.42	.24	.22
Temp. at base of smokepipe, deg. F.	485.3	499.6	557.1	553.5

* As determined from indicator cards.

† At 11.02 A. M. cut offs changed as follows: first I.P., .705; second I.P., 671; L.P., .575.

At Duluth, while loading with iron ore, the baffle marked C–D (plate III) was removed from both boilers, and an additional fire-brick baffle laid on the upper row of 4-inch tubes, as shown on A–B (plate IV). Under the changed conditions, it will be noted (plate IV) that the products of combustion no longer circulated around the superheater tubes. By opening the communicating valves between drums and main steam line, and closing the valves to the superheater, saturated steam

is thus sent to the engine. Tests III and IV were made under these conditions on the down-trip of the vessel.

With the superheaters in use, the temperature of the steam was taken just after leaving each boiler, and again near the throttle of the engine. These temperatures will be found recorded in the tables. The results of Test I show an average superheat at the boilers of 86.9 degrees and at the engine of 81.7 degrees. These figures, for Test II, become respectively 92.1 and 88 degrees.

To determine the moisture, during the tests with saturated steam, a Barrus throttling calorimeter, attached to the main steam pipe near the H.P. cylinder, was used. The results indicated that the steam was practically dry.

CONCLUSIONS.

Comparing the results obtained with superheated steam (Tests I and II), it will be seen that the power developed by the main engine differs only slightly in the two tests. The cut-off in each cylinder was precisely the same in both tests, and the only variation in the auxiliaries run worthy of note was the addition, in Test II, of two fireroom blowers and two small cooling pumps. On the basis of dry coal, the saving in fuel amounts to about 8.4 per cent. in favor of Test I. This difference is scarcely attributable alone to the steam consumption of the additional auxiliaries in Test II, and the result very likely was largely influenced by furnace conditions. The inference, in any event, is plain, viz: that for moderate rates of coal consumption with the boiler installation on the *Wallace*, natural is more economical than induced draft.

In the first three tests the H.P. cut-off was set at the same point. This resulted in Test III (saturated steam), in the development of considerably more power as compared with I and II. In order to approximate to the power shown by the latter, it was necessary to shorten the cut-off of the H.P. cylinder, which was done in Test IV.

By reference to Tests I and IV, it will be seen (excluding cylinder cut-offs) that these were made under like conditions,

as nearly as possible, with the exception that in the first, superheated, and in the second, saturated, steam was used. A comparison, based on dry coal, shows a net saving in fuel, with superheated steam, amounting to 14.5 per cent. In considering this result it should be remembered that it represents the combined increased efficiency of machinery plant, the most important factor in which is the improved economy of the engine when running under superheated-steam conditions.

NOTES.

SUBMARINE BOATS AND THEIR SALVAGE.

Broadly speaking, there are two classes of accidents that can happen to a submarine boat. First, the admission of water into the interior; secondly, an explosion. Both these have their counterpart in surface vessels—namely, collision and boiler or ammunition accidents. But the confined space and small reserve of buoyancy of the submarine boat intensifies the danger of the result to the crew. And it is this, rather than the frequency of accidents, that has occasioned public comment.

In the case of admission of water, no help can be expected from watertight bulkheads; for, obviously, to keep the vessel afloat each compartment must not have a greater capacity than the equivalent of the reserve of buoyancy; but further, for such compartments to be effective, only one—namely, the center one—can be allowed the same volume as the reserve of buoyancy, since it is not only necessary to float the boat, but also to afford egress to the crew, and therefore the boat must be maintained approximately on an even keel. To do this no compartment except the midship one must exceed in capacity half the reserve buoyancy of the boat; this limitation is necessary so as to allow of a corresponding compartment, the opposite end of the boat being flooded to maintain the horizontal trim. Such subdivision would mean at least thirty compartments, and none except the one in the center of the boat could be more than $3\frac{1}{2}$ feet between the bulkheads. Of course, such a subdivision is an impossibility. It may, however, be argued that it would be well to have one or two compartments so as to confine the water to definite portions of the boat, and to leave a certain amount of space for the crew to

*Abstract of paper by Captain R. H. Bacon, R. N., D. S. O., read before the Institution of Naval Architects.

inhabit in case of accidents. This is a very reasonable contention, but before accepting it we must consider the probabilities of an accident happening, the chances of such an accident occurring in waters where salvage is possible, and, further, the chances of successful salvage to save life under ordinary conditions of sea and weather; all these must be balanced against the ever-present inconvenience in daily life of having the boat divided into these three or four compartments. Such subdivision is not merely an inconvenience as regards control and communication, but for another reason it reacts against the efficiency of the boat when actually running on service. One marked feature in practice is the way in which spaces below a certain size affect the normal mental condition of the men. To run a boat successfully, everyone inside her should be in an absolutely normal condition; the increased constriction of vision, and the necessary accuracy of handling, argue even greater coolness than in ordinary vessels. For a man's mental condition to be normal, free living space is required; too confined surroundings react on the crew, and by producing a feeling of restriction create a mental tension; hence, unless some very strong reasons exist for making living spaces below more confined than absolutely necessary, they should be as large as the boat will permit.

THE PROBABILITIES OF ACCIDENTS.

Taking first the case of water entering the boat. Two broad conditions exist when this is possible. One, where water enters a hatch; the other, when it enters through a leak.

The former is by far the more probable cause of such an accident. Of the four serious accidents whereby boats have foundered in the last few years, in fact, all modern accidents—namely, the Russian *Delfin*, the French *Farfadet*, the *A1* and the *A8*—all have been due to water entering the hatch.

The fact of the hatch being the main source of weakness is very suggestive, and most reassuring as regards the safety of the boats, since with the practical elimination of this source of danger the main cause of accidents up to the present will

be obviated; in fact, if no other remedies were taken, the attention attracted to this point should alone be sufficient to prevent accidents of this nature happening in the future. Many suggestions have been put forward for operating such a hatch automatically; but no matter requires more careful consideration than suggestions for the automatic or semi-automatic closing of vital hatches, since, if any portion of a man, either body, hands or arms, is in the way, and is caught by the hatch, the aperture can never subsequently be made watertight, and the danger to the boat is greatly intensified. The simplest method of making the hatch of a submarine boat automatic is by hinging it on the foreside, so that the pressure of the water with the boat going ahead closes it; but had this been the case in the *A8* there is little doubt that the captain, who was leaning across the tower, would have been caught, and every subsequent chance of stopping the water entering would have ended. The best method seems to be to hinge the hatch so as to open athwartships, so that with headway on the boat no direct pressure to open or close it is exerted by the water; but should the tower submerge, the lateral inrush will undoubtedly shut it. But let us for a moment analyze the chances of the water ever reaching the hatch.

The following is the broad outline of the previously-mentioned accidents, so far as we know from the general accounts available:

In the case of the *Delfin* the hatch was quite low, too large a number of men were in the boat, the ballast tanks were flooded while the hatch was open (an original and deadly error), and the boat sank deep until the water lapped over into the boat. Such an occurrence should without doubt never have happened. No structural provision could ever forestall such an accident.

In the case of the *Farfadet* the boat was apparently diving with the hatch improperly shut, and in attempting to close it fully the hatch opened.

In the case of the *A8* the boat was being run with about one-third her normal buoyancy at considerable speed when she

dived under, and her hatch was not closed by those near it. Now all these cases are of a nature that should not occur again; there is nothing inherent in submarine boats to render any of them liable to recur.

In the case of the *Delfin*, systematic drill and the most elementary care would absolutely preclude a repetition of such an accident.

With the *Farfadet*, a redesigned closing method would allow of tightening the hatch with safety.

With the *A8*, the maintenance of full buoyancy when steaming fast would have prevented the water ever approaching the hatch 11 feet above the water line. So far, therefore, we may legitimately state that these accidents are not such as in any way to destroy confidence in the general system of submarine work.

We have now to consider the question of the admission of water due to collision or grounding.

As regards the former, a collision of any magnitude must be a most serious matter; the reserve buoyancy in the boats is so small that even a small leak must be a very difficult matter to deal with. Watertight compartments have been shown to be out of the question, and therefore, at all hazards, collisions must be avoided. This being the case, it is the natural duty of a submarine boat to get out of the way of any craft that may do her damage. Such is only wisdom on her part, and with one craft determined to avoid, the chances of collision are greatly reduced. There is one point, however, greatly in favor of submarine boats—namely, the enormous strength of their structure compared with surface boats of similar tonnage; this is of the greatest importance in all minor collisions. Two cases are on record where the boats have been struck by the propeller of a ship and have been none the worse, and many cases of collision with other minor surface craft have left the submarine absolutely unaffected, whereas the surface craft have been seriously damaged.

In the particularly sad case of the *A1*, the conning tower was struck by a large ship and the hatch slightly sprung open;

the structure otherwise was undamaged. The boat then traveled the whole length under the ship's bottom, during which time sufficient water had leaked in to prevent the boat subsequently rising. A watertight hatch now fitted at the base of the tower provides a means of guarding against any moderate leakage from such damage to the conning tower. Of course, should the hull of the boat other than the conning tower be damaged anywhere above the center line, then grave danger exists, and no considerations of design can save the boat against a considerable leak in such a position. This, however, is the one and only source of real danger to a submarine boat in the event of collision; and the only reasons why it is greater in her case than in that of surface craft are the impossibility of fitting watertight compartments and the short time available before her buoyancy is destroyed. This naturally leads us on to the consideration of what the reserve of buoyancy should be in a submarine boat. It is not permitted to discuss the design of boats, but it is infringing no secret to point out that the increase of reserve of buoyancy in the surface condition means increase of skin area for the same submerged displacement, and therefore for the same submerged horsepower. Storage of electrical power is extremely weighty, being for equal radius of action over thirty times that of the surface propulsion; for this reason only it is advisable, therefore, to keep the reserve buoyancy of the boats as low as is consistent with sea-going safety. No boat can be safer at sea than a submarine boat, since it is as tight as a bottle. No increase of actual safety is obtained by small increases in the reserve of buoyancy—say, from 15 to 30 or 40 per cent.—which entail other great disadvantages without *pro rata* gain. Provided, therefore, that the advantage gained in safety in case of collision by small increases in the reserve of buoyancy is very slight, and that the reserve is such as to ensure sea-going safety, no object exists in reducing the efficiency of the boats from the fighting point of view, by increasing the surface reserve of buoyancy unnecessarily, and so decreasing their speed when submerged.

The other cause of danger, namely, grounding, is not so very great. There are two distinct conditions: the first, merely striking the bottom, which is amply provided for by practically the whole under surface of the boats being double-bottomed, as well as by the fact of the boats being buoyant, and so very considerably reducing the effect of a grazing blow. Danger of injury to the hull from this cause is really non-existent. The other possibility is of a boat sticking its nose into the mud. The sloping nature of the under side of the fore part of the boat does not render such an occurrence probable, unless the boat were proceeding at a phenomenally large inclination, and even then the leverage exerted in getting rid of ballast, combined with the action of the propeller going astern, should exercise great influence in freeing the boat.

The last possible cause of water entering is from the boat diving to too great a depth and the hull being crushed, or some of the hull fittings being unable to stand the pressure. This argues either the failure of the diving rudders, which are made as simple and as strong as possible, and should, therefore, be above suspicion, or that too much water ballast was being carried, and, therefore, that the boat would sink from want of buoyancy. This actually occurred in the *Porpoise* in America. Fortunately the boat only sank in about 120 feet, the boat being built for 100 feet pressure. She, however, had the defect of having her tanks only constructed to stand a pressure of 25 pounds on the square inch, or, roughly, about 50 feet of water; the water in them could not, therefore, be blown out, since it was suicidal to attempt to admit the air pressure necessary to expel the water. The large pressure of water on the hull caused the boat to leak considerably, the electric pumps could not work against the pressure, and it was merely a race between the hand pump and the in-leakage of water whether she was ever raised to the surface or not. Providentially, the hand pump gained the day, and the boat floated. In any boat, every tank that can possibly be put in connection with the sea should be tested to the full hull-resisting pressure. An accident such as this could not possibly

occur with moderate care or proper design. The rate of blowing water from tanks is purposely made so great that a sinking boat should be caught and made buoyant long before a dangerous depth is approached. One turn of two handles, which are always kept in hand when the boat is diving, immediately relieves the boat of a large quantity of ballast.

Of all these possible causes of boats foundering from taking in water, we may fairly claim that the only one even probable is that when the boat is injured by collision in the hull above the center line.

EXPLOSIONS.

The question of accidents from explosions inside the boat next claims our attention. Three causes of explosions only are possible, namely:

1. From an explosive mixture of petrol and air through a leak of liquid petrol into the boat.

2. From a mixture of hydrogen given off from the batteries during charge, or in discharge, if, in the case of accident, the battery is flooded with water.

3. From the explosion of a compressed-air reservoir.

As regards petrol, the main important consideration is that to cause an explosion, first a leakage of liquid petrol is necessary; and, secondly, a spark is required to ignite the mixture. A leakage can invariably be detected by smell, should it occur; but in a properly-designed system leaks should be practically non-existent.

Even with vapor in the boat, no direct danger exists, provided the boat is properly ventilated, and no switch is moved, or anything is done to cause a spark. As a matter of fact, in practice, the smell of petrol inside a boat is almost unknown. In the case of the accident to the *A5*, where a petrol explosion occurred, the cause of the leakage was a badly-packed gland of the petrol pump, the gland being screwed down metal to metal; but, in spite of one man being overcome by the petrol fumes, the main motor was started, and the sparks determined the explosion. Had the very explicit and simple

regulations provided been carried out, no accident would have occurred. The boats are particularly free from all chance of accident with the engines at work, since the volume of air used every two minutes is equivalent in volume to that of the whole air of the boat; with such enormous ventilation going on, no chance of vapor accumulating is possible. A prevalent idea is that the petrol tanks are themselves a source of danger. A moment's consideration will show that no danger from this cause can exist; since, whenever a closed vessel has petrol inside it, the enclosed air is saturated with vapor, and is therefore non-explosive. The tanks are constructed so that leakage is impossible, and many other precautions are taken so as to minimise every source of possible leakage in the petrol system. The English boats have covered over 30,000 miles under their engines, and, with the exception of one small flash in an early boat, no explosion except that in the *A5* has occurred. This of itself is fairly good evidence of the safety of petrol when moderate care is exercised. The second possible cause of an explosion is due to hydrogen given off by the batteries in charging; but, as this operation is only carried out when the boat is opened up for ventilation, no danger exists. The formation of this gas, however, has considerable bearing on the installation of the battery, and limits the extent to which it can be made thoroughly watertight, for no risks can be run of allowing an explosive mixture being formed inside an air-tight space. The explosion which occurred in the *A8* two hours after sinking was most probably due to the formation of this gas.

The third cause—namely, the failure of the air reservoirs—is merely mentioned as a possibility; no greater danger from this cause exists on board a boat than on board an ordinary ship, where such accidents are unknown. The usual periodic tests should fully suffice to ensure these reservoirs being in a thoroughly safe condition.

We may, I think, from the above short account assume that danger to the boats from explosions is really small, and not greater in comparison than the dangers which have attended

the introduction of increased boiler and gun power in the Navy as a whole.

SALVAGE.

The ploblems connected with the salvage of these boats differ from those usually met with, chiefly because the question of rapidity is, or may be, of vital importance as regards the possibility of saving life. This factor is usually absent from other salvage operations, and no question then exists about setting deliberately to work and conducting the operations upon sound principles. The danger anticipated by the old adage, "More haste, less speed," truly applies to this as well as all engineering work.

* * * * *

Their maneuver grounds are chiefly in the deeper waters, and it is therefore evident that considerations of rapid salvage are totally out of the question should foundering occur in the majority of the waters in which they work.

The suggestion, therefore, for building special salvage craft for dealing with submarine boats when the boats mainly operate in waters where rapid salvage is impossible, is one that cannot for a moment be practically considered, even should the probable frequency of accidents warrant such action, which, as a matter of fact, it does not.

The first question as regards salvage is to provide for the succor of the imprisoned crew. Hundreds of suggestions have been received on this subject; all have received careful consideration, but up to date not one has been capable of practical application. The labor of attending to the numerous suggestions is very great, but it is well worth the expenditure of trouble if only some day one single good idea is forthcoming; and really it is impossible but to be grateful to all who attempt, however crudely, to assist the work of the country.

Let us consider the two most common and plausible of these suggestions.

Before adopting any special appliance or constructional complication, suggested solely from the point of view of saving life in accidents, two considerations should be fulfilled:

first, the fighting efficiency of the boat must not be seriously affected by it, since the boats are designed primarily for fighting, and not for meeting with accidents; and, secondly, that such arrangements, when introduced, shall be efficient against all reasonable accidents that may happen.

It is simple to design a boat to be safe against any particular accident; the impossibility lies in designing one safe against every accident. The commonest device for the release of a crew from a boat is that of providing an air lock and diving chamber, and a hatch that can be opened from the bottom of the air chamber, the pressure of air preventing the water entering when the hatch is open; the crew can then, one by one, dive out and come to the surface. Supposing a sufficient space devoted to such an arrangement, there are two considerations which militate against its success. The hatch must necessarily be in the bottom of the boat; but if an accident happens, it is "even chances," even in the uncommon case where the boat is undamaged, whether the boat rests with its fore or after end on the bottom. It is therefore an even chance whether the men would, or would not, be able to use this means of egress. Again, it must be assumed that water is present in the boat, otherwise there is no reason that she should not come to the surface, and if this water is present, it is again an even chance if the compartment can be got at from the interior of the boat. But, further, even supposing this chamber could be used, we have still to consider the effect on the crew of the variation in pressure by suddenly rising to the surface from a considerable depth. It is common knowledge that under pressure the gases absorbed in the blood get compressed, and that sudden release of such pressure allows them to expand, producing large bubbles of gas in the blood, which cause breaks in its continuity; this is productive of the most serious effects on the system, so that, even if the men were able to find egress, in many cases their lives would not be saved. Since, therefore, this system cannot be safely used in deep water, not at all on a soft bottom, and unless duplicated and placed at each end of the vessel, it is useless with a

small amount of water in the boat, and quite valueless with a large amount, it is not a fitting on which it is worth wasting much valuable weight and space. Another common suggestion is to provide a detachable chamber into which the crew may climb, which can be freed, and then will float to the surface. A few figures will dispose of the practicability of this idea. Allowing $1\frac{1}{2}$ square feet of space for each man, eleven men will require $16\frac{1}{2}$ square feet. If the chamber were made cylindrical, the thickness would be $\frac{1}{4}$ inch. It would have to be ballasted to keep it upright when floating, and, moreover, must float with a certain reserve of buoyancy to allow of the top being open even in smooth water for the egress of the men. This means that the cylinder would have to be at least 6 feet in diameter and 7 feet to 8 feet high. This would be an enormous excrescence outside the boat, and greatly increase her submerged resistance, even if further fitted with fair waters to relieve its circular shape, although increasing its size. Its dimensions would quite preclude the possibility of carrying it inside. In addition, the securing arrangements and watertight doors would add considerably to its weight, and then in the end a collision—which is, after all, the most probable cause of accident—would be liable to put it out of gear. Anyone who really considers carefully the difficulties of providing reasonable methods of exit, for a crew will soon be convinced of the futility of such special arrangements. If I am betrayed into saying that some convenient form of diving dress seems to provide the greatest chance of ultimate success, I add an appeal to non-experts who have neither seen the open sea nor have any knowledge of diving dresses to spare us written suggestions on this subject.

Suppose a boat at the bottom in less than 120 feet of water, the difficulties of rapid salvage are immense. Friendly critics who look on the Channel as a smooth sheet of water would have their ideas rudely dispelled if they had to undertake practical salvage operations. The number of days when sea and swell are absent are remarkably few, the hours available, owing to tides, are scanty, and the amount of physical work

that divers are capable of doing in deep water is remarkably little. These ever-present handicapping factors, combined with the hundred and one little difficulties and delays that crop up, soon cause hours to run into days, and difficulties, not known to the outside world, cause those who have never had such work to do to wonder where the difficulty lies. But it is useless to dwell further on this matter; let us pass to the actual salvage of a boat on the bottom.

A boat on the bottom can be in one of two conditions—either with her hull intact or with it damaged. In the former case the salvage is fairly simple, and merely consists in connecting a compressed-air pipe to a socket in the hull, admitting compressed air, and blowing the water out. I must, in passing, take the opportunity of thanking the many people who have taken the trouble to write, either privately or to the press, suggesting the provision of such a fitting. But, needless to say, this has been provided from the very first inception of these boats.

If, however, the boat is damaged, she can either be lifted bodily, or the leak may be stopped and the water blown out, when she will rise of her own buoyancy, or do so with but little assistance. In the first case a considerable weight will have to be lifted, and, providing the weather conditions are favorable, no difficulty should exist; but the operation cannot be done without considerable labor. We will imagine for a moment that there is a chance of saving life in the boat, although a leak exists. In this case it will be necessary to lift the boat not merely to the surface, but sufficiently above the surface to allow of the hatch in the conning tower being opened without the danger, when the air pressure is relieved, of letting the water rise in the hull sufficiently to drown the survivors. For this purpose center-line lifting bolts on the hull are not of much use, since the purchases shackled to them will come "two blocks" before the hull is above water. Strops made by large hawsers are more convenient, since the blocks can be kept at the side as low as the middle of the boat. Again, the blocks that must be used weigh close on half a ton

each; these are most difficult for a diver to handle. I do not know if I am addressing many who have been practical divers; if not, I must ask you to take my word as an old diver as to the enormous difficulty of bringing an unwieldy weight of half a ton near to, and of shackling it on to, a fixed point, such as an eye-bolt. Bringing a hawser to the block is far easier, or even shackling the hawser to the block on the surface, and then sweeping it into place under the boat, is a far easier and more practical method. Fixed bolts, under certain conditions, such as that of a boat being very deep in the mud, may be valuable, but for rapid salvage their utility is very problematic. Another point to be kept in mind is that the strength of the lifting gear to be used should be the sum of the weight of water in the boat and any full tanks, plus the displacement she has to be lifted to; moreover, the slinging has to be distributed so as not to strain the boat locally, cause her to leak, and allow the imprisoned air to escape from the inside.

The fact of lifting the boat above water necessitates the use of more than one lifting vessel, so that the boat may rise between them. This necessitates mooring these craft in exact position before any work can be undertaken, otherwise the tide will vitiate all subsequent operations. After everything has been done that can be done on the surface, the whole rapidity of the subsequent work depends on the tides and on the weather. To attempt to shackle on blocks below, with the lighters rising and falling 2 feet or 3 feet, is impossible, and to take the weight of the lift, except on a smooth day, is equally impossible, on account of the strain that would be brought on all the gear. The sudden immersion of the lighters and the inertia of the submarine are, in a sea, bound to strain the gear many times more than the mere weight to be lifted. For all these reasons rapid salvage is almost an impossibility, however hard everyone may work. Whatever appliances are available, wind, sea and tide will always be the determining factors as regards rapidity of work.

The mere salvage of a boat, when rapidity is not essential, does not, even when in deep water, present any great difficulty. It is, in fact, easier in the case of a strong structure, like that of a submarine, than with an ordinary ship. Moreover, the peculiar shape of the hull makes it possible to sweep hawsers underneath the ends even in waters where it would be impossible to work divers.

It has been suggested from some sources that the Admiralty should keep their own salvage plant for the general use of the Navy. This is no concern of mine, and one for the Admiralty only to consider; but if I had any weight in their councils, I would most strongly advise their doing nothing of the sort. Anyone who has been practically associated with salvage work knows that success depends largely on practical experience and constant practice. The mere plant is quite a secondary consideration, and easily obtained. Moreover, in all large salvage operations special gear has to be designed and built. Such appliances cannot be kept stored. The valuable asset to salvage companies is the experience of their divers and the constant practice that the mercantile marines of the world afford them. In such commercial enterprises the Navy can never have a share; and our own service also does not, thank goodness, provide sufficient accidents to keep the men thoroughly practiced.

In conclusion, I hope that I have been able to put impartially before you the relative safety of submarine vessels compared to that of surface craft. That the danger of the work is apt to be exaggerated, I think you will all concede. That it requires constant care goes without saying; but with the exercise of such care, no apprehension need exist of a larger percentage of accidents than in other branches of the service. That with the increase in numbers of the boats accidents will occasionally occur is undoubted, for wherever large quantities of energy are stored in an easily available state danger must exist; but that this is present to an exaggerated extent in the case of submarine boats is not a fact. Nowhere is the extent of possible danger known better than among

those who man the boats, and nowhere would the idea of excessive liability to accident be more scouted.

There is, however, another class of danger that may exist, though at present it does not do so, and that is that, in the desire to avoid all risks, the efficiency of the boats may be reduced both by constructional limitations and also by curtailing their practical maneuvering by subjecting them to work less stringent than that which simulates war conditions. Much better have no boats at all than allow such limitations to creep in. At present we can safely say that all the work the boats do is up to the full requirements of war training, and that neither of the two lamentable accidents to the boats have in any way detracted from the severity of the tactical work. The sympathy of the whole country with the sad fate of those splendid men, who, being volunteers, were the pick of the service, is apt to lead to exaggeration of the real danger of the work. It is rather outside the Navy than in it that apprehension as to the safe use of the boats arises. If this paper has succeeded in putting the real facts in a more dispassionate light, I feel that the half hour you have so kindly afforded me will not have been wasted, so far as the interests of the country are concerned.—"Engineering," London.

TURBINES AND RECIPROCATING ENGINES IN THE MIDLAND RAILWAY COMPANY'S BOATS.*

For further description of these vessels see JOURNAL A. S. N. E., Vol. XVI, pages 629, 631, 713, 1023, 1339 and 1340.

In January, 1903, the Midland Railway Company decided to build four new screw steamers for their Irish and Isle of Man services, in view of the approaching completion of their new harbor at Heysham, in Morecambe Bay, Lancashire. They entrusted the designing of these steamers to my partner, Professor Biles, and myself. The method of propulsion was naturally one that came up for discussion.

The only vessels fitted with the Parsons marine turbine

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* Paper read at the summer meeting of the Institution of Naval Architects by William Gray.

system of propulsion at that time were two small passenger steamers, trading in summer on the smooth waters of the Clyde estuary. As compared with the ordinary Clyde steamers, the results obtained were highly satisfactory, although their maneuvering qualities left something to be desired. The London, Chatham and South Eastern Railway Company had a turbine steamer under construction, but it was necessary to place the contracts for the Midland steamers before the former was tried.

After careful consideration of the data available the Midland Railway Company decided to fit two of the vessels of their new fleet with reciprocating engines and two with Parsons marine turbines.

Three of the vessels—the *Antrim*, *Donegal* and *Londonderry*—were intended for the Belfast passenger and cargo trade, and the *Manxman* for the Isle of Man summer passenger traffic, although she was also fitted with portable cabins for taking up the Belfast trade in winter.

The principal dimensions of the three former vessels are as follows: Length on the water line, 330 feet; molded breadth, 42 feet; molded depth, 25 feet 6 inches. The *Manxman* is similar in form and of the same length and depth, but the molded breadth is 43 feet.

The boiler installation in all the vessels consists of two double-ended and one single-ended boiler, the principal dimensions of which are given in parallel columns:

	Double ended.	Single ended.
Number	2	1
Length	22 feet.	11 feet 6 inches.
Diameter	15 feet 7 inches.	
Number of furnaces (total)	12	3
Heating surface in each ship	12,500 square feet.	
Grate area	400 square feet.	

The working pressure in the *Antrim*, *Donegal* and *Manxman* is 200 pounds per square inch, and in the *Londonderry* 150 pounds per square inch.

Forced draft on the closed-stokehold system is fitted in

all the vessels. The fans for this purpose are driven by electric motors in the *Antrim* and *Donegal*, and in the *Londonderry* and *Manxman* they are driven by steam engines.

The engines of the *Antrim* and *Donegal* differ only in detail. They consist of two sets of the four-cylinder triple-expansion type, each driving a three-bladed propeller. The cylinders are 23 inches, 36 inches and two of 42 inches, with a 30-inch stroke.

The auxiliary machinery in all the steamers is of the most modern type, and, as the pumps are driven independently, a better comparison is afforded of the performances of the propelling machinery.

The arrangement of the turbines in the *Londonderry* and *Manxman* differs only in detail, but the turbines in the *Manxman* are larger, as they were designed for 25 per cent. more power than the *Londonderry*. The *Manxman* is also fitted with a Parsons vacuum augmentor. In each vessel there are three turbines; one high pressure and two low pressure. With the latter are incorporated the reversing turbines that work *in vacuo* when not in use. Each of the three turbines drives a separate shaft and a three-bladed propeller. The low-pressure turbines are on the outer shafts and the high-pressure turbine on the center shaft. All the turbines in both ships were made by Messrs. Parsons' Marine Steam Turbine Company.

The propellers of the *Antrim* and *Donegal* are three-bladed, the *Antrim's* being built and the *Donegal's* solid. The *Antrim's* is a little smaller in diameter and coarser in pitch.

The propellers of the *Manxman* are as follows: center, 6 feet 2 inches in diameter, 5 feet 7 inches pitch; side, 5 feet 7 inches in diameter, 5 feet pitch. The *Londonderry's* are all the same—5 feet in diameter, 4 feet 6 inches pitch.

The conditions of the contract were that each vessel was to maintain a speed of 20 knots for six continuous hours with the double-ended boilers only under steam and 300 tons dead weight on board.

The results of the official trials were as follows:

	Antrim.	*Londonderry.*	*Manxman.*
Speed in knots.....	20.6	21.6	22.65

With all the boilers in use the results were:

	Antrim.	*Londonderry.*	*Manxman.*
Speed in knots.....	21.86	22.36	23.12

The amount of water consumed was measured during the progressive trials by counting the strokes of the feed pumps, and the comparative results are given in the following table:

Speed in knots.	Gallons of water consumed per hour.		
	Antrim and *Donegal.*	*Londonderry.*	*Manxman.*
14	4,500	4,500	4,500
17	6,700	6,100	5,800
20	9,700	8,900	8,300
22	...	13,600	12,500
23	..	...	17,300

These figures throw some light on the relative economy not only of the two systems of propulsion at various speeds, but on the different arrangements in the two turbine steamers themselves. They show that at from 14 knots to 20 knots the turbine is more economical.

The maximum difference occurs between 19 and 20 knots, which is the working speed of the vessels on service. In the case of the *Londonderry* the decrease in water consumption amounted to 8 per cent., and in the case of the *Manxman* to 14 per cent., as compared with the *Antrim* and *Donegal.*

The service to and from Heysham and Belfast was opened on September 1, 1904, and since that date there has been one vessel plying each way every night except Sunday. Since May 1, 1905, the *Manxman* has been employed exclusively on the Isle of Man route; and the results, being obtained under different conditions, are not included in the comparisons.

Through the courtesy of the officials of the Midland Railway Company the log books of the four steamers have been placed at my disposal, and it is, therefore, possible to make

some comparisons that may be of interest to those connected with ships. As the question under consideration is reciprocating engines *versus* turbines, those occasions when the *Antrim* and *Donegal* were running together have been omitted. The *Londonderry* and the *Manxman* have never been running together on this service.

An accident to the high-pressure turbine of the *Londonderry* that happened early in the month of November interrupts the records of that steamer for three months. The cause of the accident has never been definitely ascertained, but everyone is agreed that no fault can be attributed to the turbine as a means of propulsion. The dummy rings on the casing and drum were allowed to come into contact, and one-half of them were destroyed, but the circumstances were rather peculiar, and not likely to occur in normal conditions of working.

Table A gives the following particulars for the various steamers, viz:

The number of single trips made from Belfast to Heysham, and *vice versa*.

The number of tons of coal consumed while under steam.

The total number of hours under steam.

The total number of hours at full speed.

The amount of coal consumed in port varies according to circumstances, and as it does not materially affect the matter under discussion, it has been excluded from these tables. It amounts approximately to about eight tons per steamer per single trip.

The column "Hours under steam" gives the time from quay to quay, and includes the time occupied in maneuvering the vessel in harbor.

The table treats the matter from a purely commercial standpoint; that is to say, it ignores the differences of speed and coal consumption due to the relative weights of turbines and reciprocating engines, and only deals with the items that exclusively interest the shipowner. It gives the amount of coal that each vessel consumed on a given number of trips. In all four steamers the three boilers have been in use, and all have

Table A.

Dates.	Number of passages.	T.-S.S. *Antrim.* (Reciprocating engines.)			T.-S.S. *Donegal.* (Reciprocating engines.)			*Londonderry.* (Turbines.)			*Manxman.* (Turbines.)		
		Tons of coal consumed.	Hours under steam.	Hours full speed.	Tons of coal consumed.	Hours under steam.	Hours full speed.	Tons of coal consumed.	Hours under steam.	Hours full speed.	Tons of coal consumed.	Hours under steam.	Hours full speed.
1904.													
Sept. 1 to 10.....	9	282	67.3	55.1	...	...	...	284.8	65.3	54.9	...	...	...
23 to 28.....	5	159.5	36.3	28.1	...	...	...	161	34.3	30.5	...	...	...
29 to Oct. 17..	16	...	...	...	498.4	116.5	103.3	527.4	111	97.5	...	...	...
Oct. 18 to Nov. 1...	13	447.5	89	71	...	...	...	429.3	92.2	75	...	...	...
Dec. 1 to 15.....	13	...	...	...	465	91.3	78.5	...	...	...	507.7	84.1	74.1
16 to 21.....	5	204.7	34	32.1	...	...	...	...	...	...	187	39.1	23.5
1905.													
Dec. 29 to Jan. 10..	11	416.4	73.1	67	...	...	...	...	...	...	420.5	73.3	62.5
Jan. 25 to Feb. 8...	13	...	...	...	520	89.8	76	...	...	...	532	92.1	59.6
Feb. 9 to 23.....	13	499.7	89	72.1	...	...	...	...	...	...	512	83.8	73.5
Mar. 11 to 25.....	13	...	...	...	522	86.8	77.2	...	...	...	530	86.2	70.7
31 to April 10..	9	345.6	58	52.5	...	...	...	350.5	62.5	52.2	...	...	...
April 26 to May 10..	13	...	...	...	551	89.2	78.4	512	84	72.4	...	...	...
May 12 to 25.....	12	472.8	74.5	68.2	...	...	...	470.3	77.4	67.5	...	...	...
June 10 to 24.....	13	...	...	...	459.1	87.1	78.5	510.5	82.9	72.9	...	...	...
	158	2,828.2	521.2	446.1	3,015.5	560.7	491.9	3,245.8	609.6	522.9	2,689.2	458.6	363.9

SUMMARY.

	Antrim.	*Donegal.*	*Londonderry.*	*Manxman.*
Number of passages.....	77	81	90	68
Coal per passage, tons.....	36.7	37.2	36.1	39.6
Hours at full speed per passage.....	5.78	6.07	5.81	5.35
Hours at full speed, percentage of total hours under steam.....	85.5	87.7	85.7	79.5

been working with the same moderate air pressure in the stokeholds. The results, therefore, represent the amount of coal consumed under easy steaming in all circumstances.

To show the results obtained from each pair of steamers running simultaneously, but in opposite directions, the previous table has been split up, as shown in Table B.

Table B.—Showing Results Obtained by Steamers Running Simultaneously, but in Opposite Directions.

	Reciprocating engine.	Turbine.
	Antrim.	*Londonderry.*
Number of trips	48	48
Average coal per trip, tons	35.6	35.3
speed in knots	19.7	19.5
	Donegal.	*Londonderry.*
Number of trips	42	42
Average coal per trip, tons	36	36.9
speed in knots	19.2	19.8
	Antrim.	*Manxman.*
Number of trips	29	29
Average coal per trip, tons	38.6	38.6
speed in knots	19.5	20 3
	Donegal.	*Manxman.*
Number of trips	39	39
Average coal per trip, tons	38.7	40.2
speed in knots	19.3	20.3

The speeds obtained on the trial trips have been already dealt with, but, valuable as these results are, the conditions in which trial trips are conducted are more or less artificial, and shipowners are justified in not taking them as final guides on the subject.

The logs have been very carefully examined, and, neglecting those runs where full speed was not maintained for the whole time that the vessels were in the open sea, the results are as given in Table B.

These results point to a marked decrease in the coal consumption of the *Manxman*, as compared with the *Antrim* and *Donegal*. The *Manxman* did 20.3 knots for the same coal consumption that the *Antrim* had at 19.5 knots. A similar comparison of the *Manxman* with the *Donegal* gives nearly the same result.

In other words, for a speed of 19.5 knots the *Antrim* requires 38.6 tons of coal, and the *Manxman* 35 tons—a saving of 9.3 per cent. The *Donegal*, for a speed of 19.3 knots, requires 38.7 tons, and the *Manxman* 35.4 tons—a saving of 8.5 per cent.

The performances of the *Londonderry* are nearly as efficient as those of the *Antrim*, but they are better than those of the *Donegal*.

They also indicate that the *Manxman*, with higher steam pressure, a smaller number of revolutions, and larger propellers, has done better than the *Londonderry*.

A further economy in the turbine steamers is effected in the amount of oil used for lubrication. The logs show that this amounts in both steamers to five gallons per single trip. This, again, permits of a further economy in the reduction of the engine-room staff from four greasers to two.

Speaking generally, therefore, the performances of the turbine steamers, especially the *Manxman*, have been greatly superior to those of the steamers fitted with reciprocating engines.

It is not possible to make a quantitative analysis of the cost of upkeep, but so far the turbines have cost practically nothing (excepting the cost of repairs due to the accident to the *Londonderry*), and they require very little attention compared with what is necessary in the very best running engines of the reciprocating type.

There can be no doubt that one great benefit derived from the use of turbines is the elimination of the vibration inseparable from the use of reciprocating engines in Channel steamers. The engines in the *Antrim* and *Donegal* were most carefully balanced, and the vibration at the service speed is almost imperceptible. The almost unanimous testimony, however, of those who have traveled in the vessels is that the turbine steamers are steadier.

The only real inferiority in the *Londonderry* and *Manxman* is the difficulty of maneuvering from rest in narrow waters. In this respect they compare unfavorably with the *Antrim* and

Donegal. No doubt with further experience those in charge will become more expert in working with the small screws, but for turning from rest there is smaller power in the side propellers. Experiments were made at the trial trips that showed that the turbine steamers going full speed could be brought to rest in about a minute and a half. This is a good result, but actual experience has shown the relative inadequacy of the backing power starting from rest. There is no good reason why sufficient backing power cannot be obtained with small screws if the reversing turbines are made powerful enough.

Having dealt with the question of speed and coal consumption, there remain three other points in connection with the design, viz: space occupied, weight and cost.

The turbines occupied so much more floor space than the reciprocating engines that the electric plant had to be put in the tunnel.

Space may be saved on the upper decks, but this is not an unmixed advantage, as it was effected by reducing the light and air space to the turbine room. By our tonnage laws as at present, the actual volume of the light and air space is included in the gross tonnage; but $1\frac{3}{4}$ times the actual volume may be deducted in arriving at the net tonnage. The effect of decreasing the light and air space to the turbine room involved an increase in the net tonnage of nearly 50 tons.

	Gross tonnage.	Net tonnage.
Antrim	2,100	603
Londonderry	2,086	651

In the case of vessels trading between ports where tonnage dues are heavy this is a matter of great importance, and may balance the commercial gains in oil and fuel. In such cases the gain in space cannot be used with advantage.

The saving of weight in the turbine steamers is considerable. In the hull the weights involved, viz: engine seat, tunnel stools, boss frames and plating, &c., are reduced by 30 tons. Against this the buoyancy of the bosses is 29 tons less in the turbine than in the other steamers, so that from the draught

point of view the saving in hull is neutralized by the loss of displacement.

The weight of engines, shafting and propellers in the *Antrim* and *Donegal* was 280 tons. The weight of the corresponding items in the *Manxman* was 195 tons, so that the total weight to be propelled is about 115 tons less, a difference of nearly 6 per cent. on the light weight of the steamer.

The difference of the initial cost of the turbines as compared with reciprocating engines is not great. Comparing the *Antrim* and the *Londonderry*, it amounted to 1½ per cent. of the total cost of hull and machinery.—"Engineering," London.

GAS ENGINE WITH PRODUCER FOR MARINE USE.

The following is an extract from an article in "Engineering," London, descriptive of the installation manufactured by Messrs. John I. Thornycroft & Co., Limited, London:

The vessel is built of steel, and is 60 feet long, 10 feet wide, and 2 feet 6 inches draught. She is a single-screw boat, the propeller working in a tunnel. There is a roomy cabin forward and another aft of the machinery space. Behind this again is a large watertight well, the steel floor of which is above the water level. There are self-acting freeing scuppers with hinge valves, on the life-boat principle. The fore peak is divided off by a watertight bulkhead, and the machinery space is between watertight bulkheads. Lavatory accommodation and other conveniences are provided.

The interesting feature of the vessel is naturally the novel type of machinery, which consists of a four-cylinder Thornycroft gas engine, on the Capitaine system. The cylinders are single acting, each 8¼ inches in diameter and 11 inches stroke, and develop collectively 75 brake horsepower at 300 revolutions per minute. In regard to general construction, the framing consists of mild steel plates connected by angle bars, thus forming a box-like structure of great rigidity, the steel plates being carried right round to form the crank pit. Each cylinder trunk is cast separately, and is contained in the fram-

ing. The cylinder heads or combustion chambers are also separate castings, and are supported between plates riveted to the transverse members of the frame. Each cylinder trunk is attached to the head by means of four bolts. In this way the whole force of the explosion is taken by means of the plates riveted to the transverse members, and transmitted directly to the crank-shaft bearings, the latter being also bolted to the frame plates. The object aimed at is to do away with heavy bolts that would otherwise be needed to connect the cylinder head to the trunk, such as are usual with the ordinary construction of engines of this description. The cylinder-head joint is not, with this arrangement, likely to leak—a frequent source of trouble with the ordinary arrangement. Another advantage is that the cylinder head, which carries all valves and the ignition apparatus, may be readily removed for cleaning and inspection; the operation of disconnecting and replacing can be carried out in as short a time as six minutes. The importance of this feature will be appreciated by those who have had experience with suction gas plants.

All valves are actuated mechanically by means of cams, the cams being actuated by toothed gearing from the crank shaft. The valves open directly into the combustion chamber, thus minimizing the surface which would cool the gases. The cam shaft is hollow, and carries the spindle for actuating the ignition gear. This spindle is under the control of the governor, so that the spark is advanced or retarded according to the speed of engine required, the governor spring being hand-controlled at will. The governor also acts on a valve for throttling the mixture. The ignition is effected by a low-tension magneto, the magneto being on the after part of the engine frame, and driven by bevel gear from the governor shaft. The bevel wheel for driving the magneto is also controlled by the governor, so that the time of maximum current synchronizes with the time of ignition. A distributor is fitted, so that should one of the ignition plugs become short-circuited the working of the others would not be interrupted.

The cylinder heads are cast with jackets for water circula-

tion, the cylinder trunks being fitted with a thin steel casing, water circulating between the cylinder and the trunk. Each cylinder head and each trunk has its separate service, which is at atmospheric pressure. By this arrangement it is possible to see whether the circulation is being properly maintained. The cooling water is circulated by a centrifugal pump, driven from the gas drier by a belt. There is forced lubrication for the cylinders, the bearings being lubricated by drip feed. There is an oil plunger pump, driven from the crank shaft, and this delivers oil to a mechanical distributor having eight pressure-feed connections and nine sight-feed connections. The gudgeon pins of the connecting rods are hollow, and serve as oil retainers, the oil being supplied by means of a syringe. This system of lubrication is economical, and has been found to act with certainty.

For starting the main engine an auxiliary petrol motor is used. This small engine is mounted on a pivoted base plate, and is swung over to tighten the belt when required to be used. When the main has been started, the belt of the starting engine is thrown off. Half-compression cams are fitted to the main engine to ease the starting. They are brought into play by hand, and released by the governor when the engine has attained its normal speed. Reversing is effected by means of epicyclic gear and a cone clutch, placed in the line of shafting, and forward of the thrust block. One hand wheel frees and reverses the engine in either direction.

The suction producer consists generally of a steel shell having a fire-brick lining surrounding the hot zone. The upper part forms a hopper for holding a supply of fuel, which descends automatically, as required. In the boat in question the supply will last for ten hours without recharging. Anthracite or coke is used. The fire bars are of channel section, to enable them to hold ash and to better withstand the intense heat. They are supported on cams, and may be lowered towards either side to facilitate clinkering. A steam generator is placed in the upper part of the producer. This consists of a shell with Field tubes, and serves the double

purpose of cooling the cases and generating steam, which is decomposed in the fire in order to supply the necessary gases for the explosive mixture. It should be explained, however, that there is an alternative means of generating steam by the exhaust gases from the engine, and this is fitted to the boat in question. The gas is passed from the producer into a cooling tower, there being a spray of water from the top which falls by gravity; and there is also a finely-divided water jet injected into the bottom by means of compressed air. This serves to free the gases from dust or other objectionable matter.

For the effective working of the engine it is necessary that the gases should not contain moisture; and to free them from water a centrifugal dryer is fitted. The dryer also removes any trace of tar that may have escaped through the cooler. The drier runs at a high rate of revolutions, and is driven by a belt from the engine crank shaft. It forms an essential part of the apparatus, and is a distinguishing feature of the system. Two small centrifugal pumps are connected to the drier spindle. One of these, to which reference has already been made, is for supplying cooling water to the engine and gas cooler, the other being for drawing off the dirty water from the cooler and discharging it overboard. For mixing the air and gas there is a double-seated valve situated in the inlet pipe near the engine, air and gas being automatically regulated to the proper mixture to suit the speed of the engine; but for different qualities of gas the air regulation is by hand.

Preliminary trials have already been made with this system, and it has been found to give satisfactory results.

COMPARATIVE LOSSES IN THE RUSSO-JAPANESE WAR.

* * * * *

According to the figures given by the "Times" correspondent up to the date he states, Russia has lost 346,588 tons of fighting ships, not including minor craft; while the Japanese Navy will emerge from the fight with a displacement of 25,435 tons greater than when the war began. This latter figure, however, must be largely increased, as the latest information from

Japan shows that a number of the large Russian vessels which were sunk have been raised, and they will be added to the Japanese Navy. The story of the Russian naval squadrons in Far Eastern seas since the war commenced is shown in the following table:

RUSSIAN NAVY.

Nature of vessels.	Engaged in the war.		Sunk.		Captured.		Interned.		Remaining.	
	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.
		tons.		*tons.*		*tons.*		*tons.*		*tons.*
Battleships	15	180,980	12	144,958	2	23,110	1	12,912	...	...
Armored cruisers	7	63,533	5	38,979	...	...	...	...	2	24,554
Sea-going coast defence ironclads	3	13,212	1	4,126	2	9,086	...	...	...	...
Cruisers	13	65,416	6	26,341	...	...	5	29,115	2	9,960
Other kinds, excluding converted cruisers	21	76,793	14	25,222	1	11,700	2	11,449	4	28,422
Destroyers	24	10,290	19	5,666	2	590	11	3,334	2	700
Totals	83	410,224	57	245,292	7	44,486	19	56,810	10	63,636

This table is not quite complete, as it does not include torpedo boats, special-service steamers, as well as converted cruisers and a number of vessels whose fate is uncertain. The broad result is very striking: out of a total of 83 ships, with a displacement of 410,224 tons, sent by Russia into the belligerent arena, only ten, with a displacement of 63,636 tons, remain in her fighting line.

The corresponding table for the Japanese Navy is to be found on page 879.

This table also does not include torpedo boats, special-service steamers and converted cruisers. A comparison of the two tables brings out the fact that the Japanese force of 76 vessels, with a displacement of 274,184 tons, has sunk or captured 64 ships, with a displacement of 289,778 tons. On the other hand, while the Japanese have lost 12 ships, with a total displacement of 46,025 tons, they have captured or raised more than double this amount of tonnage. From the news-

papers received from Japan by last mail we see that the Japanese naval authorities call attention to the fact that the *Peresviet*, which has just been raised at Port Arthur, was one of the strongest ships in Russia's Pacific Squadron. She left Kronstadt in 1901.

JAPANESE NAVY.

Nature of vessels.	Engaged in the war.		Sunk.		Captured.		Interned.		Remaining.	
	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.
		tons.		*tons.*		*tons.*		*tons.*		*tons.*
Battleships	6	86,299	2	27,757	...	...	...	...	4	58,542
Armored cruisers	8	74,178	...	...	...	...	...	...	8	74,178
Sea-going coast defence cruisers	2	11,112	...	...	...	...	...	...	2	11,112
Cruisers	20	68,481	4	12,733	...	...	...	...	16	55,748
Other kinds	18	26,689	4	4,797	...	...	...	...	14	21,892
Destroyers	22	7,425	2	738	...	...	...	...	20	6,687
Totals	76	274,184	12	46,025	...	...	...	...	64	228,159

The *Peresviet* and the *Ossliabya* were laid down in 1898, under Russia's new program of naval expansion, and were supposed to embody the best results of modern naval progress. The approval that was accorded to them by expert opinion led to the laying down of the *Pobieda*, a sister ship, in 1899. These vessels did not carry a very heavy armament—four 10-inch and eleven 6-inch guns. The "Hochi Shimbun" says that the *Poltava* will be the next vessel to be floated, and that she will be followed by the *Pobieda*. The *Poltava* is a smaller type of vessel, and somewhat older. Her displacement is 10,960 tons, and her speed only 16 knots. She is lying on an even keel. The *Pobieda* has a slight list, but it is not expected that any serious difficulty will attend her raising. The *Retvisan* (battleship, 12,700 tons, completed in 1900) and the *Pallada* (protected cruiser, 6,630 tons) are said to be the most injured, but the expectation is that they too will be saved, and in that event the *Sevastopol* alone would remain at the bottom. This forecast, if trustworthy, means that there will be added to the Japanese Navy three first-class and one second-class battle-

ships, one first-class armored cruiser, and one protected cruiser. Adding the results of the battle in the Sea of Japan, and the salvage operations at Chemulpo, we get this table :—

SHIPS TO BE ADDED TO JAPANESE NAVY.

	Tons.
Orel, battleship, 1902, ...	13,516
Retvisan, battleship, 1900, . . .	12,700
Peresviet, battleship, 1898, . . .	12,674
Pobieda, battleship, 1900, . . .	12,674
Poltava, battleship, 1894, . . .	10,950
Nicholai I, battleship, 1889, . . .	9,700
Bayan, armored cruiser, 1900, . .	7,800
Varyag, protected cruiser, 1899, . .	6,500
Pallada, protected cruiser, 1899, . .	6,630
Seniavine, coast-defence ironclad, 1894, .	4,126
Apraxin, coast-defence ironclad, 1896, .	4,126

It is stated that much of the success achieved in raising these ships is due to a new pumping machine recently imported. Two of these are at work. Each pumps 4,000 tons per hour. The name of Captain Sakamoto is specially mentioned in connection with the work. He commanded the unfortunate *Yashima*, and his exceptional skill in salvage operations has been conspicuous in the raising of the *Bayan*. These facts and figures speak volumes for the efficiency of the Japanese Navy, while the report of the Russian admiral reveals a state of affairs, both as regards men and material, which go a long way towards explaining the complete defeat of the Russian squadrons. The "Kokumiu Shimbun" remarks truly :— "Never was a Navy burdened with a heavier and more onerous responsibilty than the Imperial Japanese Navy at the beginning of the war. The enemy's Vladivostock and Port Arthur squadrons combined were in themselves a fair match for all the naval strength we could place under Admiral Togo's command; but, in addition, the enemy had the Baltic fleet to draw upon, while we had nothing more to fall back upon ; so that Admiral Togo's work was not only to demolish a strong foe present—

a task hazardous enough in itself—but to exterminate his reinforcements to come. How well Admiral Togo executed his part the results very clearly show."

The war has also been the means of adding considerably to the Japanese mercantile marine. The appended list gives the

Ship's name.	Nationality.	Tonnage.
1. *Ekaterinoslav*	Russia	5,627
2. *Mukden*	Russia	1,567
3. *Russia*	Russia	2,312
4. *Argun*	Russia	2,458
5. *Manjuria*	Russia	6,193
6. *Alexander*	Russia	261
7. *Resnick*, sailing ship	Russia	87
8. *Nicolai*	Russia	123
9. *Mikhael*	Russia	3,461
10. *Kotic*	Russia	400
11. *Juliade*	Russia	?
12. *Manchuria*	Russia	2,937
13. *Bobrick*, sailing ship	Russia	125
14. *Nadejuta*	Russia	68
15. *Thalia*	Russia	?
16. *George*	France	179
17. *Fuhping*	Germany	1,393
18. *Veteran*	Germany	1,199
19. *Nigretia*	England	2,368
20. *King Arthur*	England	1,416
21. *Rosalie*	England	4,370
22. *Redington*	England	4,421
23. *Wilhelmina*	Holland	4,295
24. *Bawlry*	England	2,407
25. *Oakley*	England	3,798
26. *Burmah*	Austria	3,071
27. *M. S. Dollar*	America	4,216
28. *Wyefield*	England	3,235
29. *Siam*	Austria	3,160
30. *Pulos*	Germany	2,398
31. *Apollo*	England	3,829
32. *Scotsman*	England	1,677
33. *Silvania*	England	4,187
34. *Powderham*	England	3,019
35. *Severus*	Germany	3,307
36. *Romulus*	Germany	2,630
37. *Easby Abbey*	England	2,963
38. *Vegga*	Sweden	2,562
39. *Venus*	England	3,558
40. *Aphrodite*	England	3,949
41. *Tacoma*	America	2,812
42. *Harbarton*	England	3,256
43. *Industrie*	Germany	198
44. *Henry Bolckow*	Norway	1,006
45. *Quang Nam*	France	1,431
Total	over...	108,000

merchant ships captured by the Japanese Navy, and no doubt it will interest many of our readers. When not described as a sailing ship, steamer is to be understood. It is to be noted that in no case was a non-combatant vessel sunk by the Japanese Navy.

The Russian record of vessels captured is very different from that of the Japanese. It is not necessary to reproduce it, as it only contains the names of fifteen ships, which have an average tonnage of 234 tons, only one being over 1,000 tons.

* * * * *

—"Engineering," London.

TEST OF CURTIS STEAM TURBINE.

The data given below was obtained from the test of a 2,000-kilowatt Curtis turbine (900 revolutions per minute), made by Mr. Fredrick Sargent, of the firm of Sargent & Lundy, engineers, and Mr. Louis a Ferguson, vice-president and general manager of the Commonwealth Electric Company.

The machine is of the General Electric Company's standard four-stage type, designed in 1903, with a few changes, made as the result of experiments conducted during the past year. Other changes, to be embodied in future machines, but which could not be made in the one under test without rebuilding, are expected to give an even higher efficiency, and there seems to be no doubt but that the economy of these machines will equal that of any of the best European types.

On the strength of this test the Commonwealth Electric Company placed an order with the General Electric Company for two turbo-generators, each with a nominal capacity of 8,000 kilowatts and a maximum short-period capacity of 12,000 kilowatts.

All instruments used were carefully tested and standardized, and the surface condenser used showed practically no leakage.

Several preliminary tests gave results closely approximating those of the official one.

DATA FROM TEST OF 2,000-KILOWATT CURTIS STEAM TURBINE, SCHENECTADY, N. Y., MAY 3, 1905.

FULL-LOAD TEST.

Duration of test, hours	1.25
Steam pressure (gauge), pounds	166.3
Back pressure (absolute), inches of mercury	1.49
Superheat, degrees Fahrenheit	207
Load, in kilowatts	2,023.7
Steam consumption per kw. hour, pounds	15.02

HALF-LOAD TEST.

Duration of test, hour	0.916
Steam pressure (gauge), pounds	170.2
Back pressure (absolute), inches of mercury	1.40
Superheat, degrees Fahrenheit	120
Load, in kilowatts	1,066.7
Steam consumption per kw. hour, pounds	16.31

QUARTER-LOAD TEST.

Duration of test, hour	1
Steam pressure (gauge), pounds	155.5
Back pressure (absolute), inches of mercury	1.45
Superheat, degrees Fahrenheit	204
Load, in kilowatts	555
Steam consumption per kw. hour	18.09

ZERO LOAD.

Duration of test, hours	1.33
Steam pressure, pounds	154.5
Back pressure (absolute), inches of mercury	1.85
Superheat, degrees Fahrenheit	156
Steam consumption per hour, pounds	1,510.5

In addition to the data obtained from the test made by Messrs. Sargent and Ferguson the following water rates were obtained for the same machine, running at 900 revolutions per minute and 175 pounds absolute pressure, under different conditions of superheat and vacuum:

Saturated	27½ inches	Water rate = 18.6
100 degrees superheat	27½ inches	Water rate = 17.15
150 degrees superheat	27½ inches	Water rate = 16.6
200 degrees superheat	27½ inches	Water rate = 16.1
Saturated	28 inches	Water rate = 18.1
100 degrees superheat	28 inches	Water rate = 16.7
150 degrees superheat	28 inches	Water rate = 16.15
200 degrees superheat	28 inches	Water rate = 15.65
Saturated	28.5 inches	Water rate = 17.6
100 degrees superheat	28.5 inches	Water rate = 16.3
150 degrees superheat	28.5 inches	Water rate = 15.75
200 degrees superheat	28.5 inches	Water rate = 15.25

At 150 pounds absolute takes 103.5 per cent of these values.
At 200 pounds absolute takes 97.5 per cent.
At 215 pounds absolute takes 96 per cent.
At one-half load the water rate is approximately one pound higher.

LAP JOINTS AND BOILER EXPLOSIONS.

This journal has so frequently expressed its opinion that the most dangerous features in engineering works are generally the minor details, that it is hardly necessary to elaborate on the subject generally at this time. A specific instance of this truth is afforded, however, by the final report on an investigation of the Brockton boiler explosion, made a little while ago by the Hartford Steam Boiler Inspection and Insurance Co. The preliminary report of the company on this very disastrous accident was printed in the Current News Supplement of this journal on April 1, and it was then stated that the explosion was probably caused by a lap-joint crack. The boiler was frequently and carefully examined, yet it blew up on the day after it was placed in service, after an idle period, while it was carrying only its regular pressure of 80 pounds. It was 6 feet in diameter and 17 feet long, and built of $\frac{3}{8}$-inch fire-box steel, made by Carnegie, Phipps & Co. The thorough examination made by the company's experts has led them to confirm their earlier statement of the cause of the explosion, and as the lap joint is used in about seven-eighths of the boilers now in service, it is manifest that it is a very important detail.

It is well known that in the passage of a plate through the bending rolls, for the purpose of giving it the proper form for a boiler, a great deal of strain in the metal is inevitably caused. If this were not the case the sheet would not be curved after the process, for steel is sufficiently elastic to resume its original form after being subjected to large stresses. Another fact that can be observed at any set of bending rolls is the lack of uniform curvature in a solid sheet at those edges where it enters and leaves the rolls. If these edges have been punched, the plate is often bent considerably along one of the lines of rivet holes. It is these edges which are brought together and

riveted, and it is often necessary to force them together by sledging or its equivalent. In the case of a lap joint, the extra strains coming on the metal do not cease with the completion of the manufacturing processes, for when the boiler is put in service the tension in the two plates has a tendency to twist the joint in a way perfectly apparent to the reader who will make a sketch of such a joint. For all these reasons it is manifest that the steel used in a boiler must be selected with full regard for its ability to withstand these special stresses, due to the use of the lap joint, and the necessity for marked ductility in the plates, particularly those of considerable thickness for shells of small diameter, needs no demonstration.

Even with the best of care and materials a dangerous defect is liable to occur in the lap joint, which has received the name of the lap-joint crack. It was such a defect that caused the loss of 58 lives, injuries to 117 more people and the destruction of property valued at about a quarter of a million dollars. With such facts before one it is almost criminal to look upon the design of boiler details other than with extremely critical eyes, or to fail to study this peculiar class of cracks.

The lap-joint crack is defined by the Hartford company as a fracture of one of the plates, which follows the general course of the joint under the extreme edges of the rivet heads in such a position as to be entirely covered by the projecting lap of the unaffected plate. They have been discovered many times in this country and abroad, and in Great Britain some of the leading authorities on boiler construction have drawn attention specifically to their dangerous nature. The main fact to observe is that the crack, no matter which plate it may be in, always starts from that face of the affected plate which is in contact with the overlapping plate, progressing into the metal more and more deeply until the boiler is weakened perhaps to the point of explosion. It cannot be seen from either the outside or inside of the shell until it has penetrated clear through the metal or has run into a rivet hole, so that its presence is shown by escaping steam or minute cracks. Before such indications can be observed the boiler may have reached a condition of extreme

danger. The inspectors of the Hartford company have found examples of such cracks running for long distances along the joint without perforating the metal at any point, and Mr. Fletcher, of the Manchester Steam Users' Association, has made a somewhat similar assertion concerning an examination he once made.

This danger has been so well known that some boiler designers and makers are urging the passage of legislation forbidding the use of boilers having lap joints. They assert that no boiler having butt joints has ever exploded, but neglect to add that the number of these boilers is very small compared with the other type. It is a question whether such a legal prohibition is advisable. A well-made lap joint in a boiler constructed of suitable steel is not essentially dangerous, although the result is by no means so good as a boiler of equal quality with butt joints. Instead of prohibiting the use of the lap joint it would be wiser for the present to hedge its use about with so many precautions that it will become as safe as such construction can be made.—"Engineering Record."

SHIPS.

UNITED STATES.

Kansas—*Launch of.*—The U. S. battleship *Kansas* was successfully launched from the yard of the New York Shipbuilding Company, Camden, N. J., at noon on August 12th.

The construction of the *Kansas*, together with her sister ships, the *Vermont* and *Minnesota*, was authorized by an act of Congress in 1903.

The general characteristics of these vessels are as follows: Trial displacement, 16,000 tons; mean draught for trial displacement, 24 feet 6 inches; length on load-water line, 450 feet; length over all, 456 feet 4 inches; breadth, extreme, 76 feet 10 inches; speed, 18 knots.

The main battery will consist of four 12-inch, eight 8-inch, and twelve 7-inch breech-loading rifles. In the secondary battery there will be twenty 3-inch rapid-fire guns of 50 calibers in length, twelve 3-pounder semi-automatic, six 1-pounder automatic, two 1-pounder semi-automatic guns, two 3-inch field pieces, two machine guns of .30 caliber, and six automatic guns of .30 caliber.

For armor protection there will be a complete water line belt 9 feet 3 inches wide and 9 inches thick for a distance of about 285 feet amidships, with a reduction to 4 inches at the bow and stern; the armor for the lower casemate is to be 7 inches thick for the sides and 6 inches athwartships, while for the upper casemate the thickness is 7 inches for both sides and athwartships.

For the 12-inch guns the turret armor is 12 inches at front, 8 inches on sides and rear, and $2\frac{1}{2}$ inches on top; that for the barbettes is 10 inches at front and $7\frac{1}{2}$ inches at rear above the gun deck, and 6 inches throughout between the gun and protective decks.

In the case of the 8-inch guns, the turret armor is 6½ inches at front, 6 inches at sides and rear and 2 inches on top; the barbette, 6 inches front and 4 inches rear.

For the conning tower and shield the thickness is 9 inches throughout, with 2-inch top plates, and a 6-inch central tube having an internal diameter of 36 inches.

The protective deck will be of two courses, the lower one of 20-pound ship steel and the upper of 40-pound nickel-steel on the flat and 100-pound nickel-steel on the slopes and at ends.

A cofferdam about 30 inches wide will extend aft from the bow to within 16 feet of the stern.

The propelling machinery will consist of two vertical, inverted, 4-cylinder, direct-acting, triple-expansion engines, designed to develop 16,500 horsepower when running at 120 revolutions per minute with a steam pressure of 250 pounds in the high-pressure cylinder.

The cylinder diameters are 32½, 53 and (two) 61 inches, with a common piston stroke of 48 inches.

The boilers, of the Babcock & Wilcox type, twelve in number, have a total grate surface of 1,100 square feet, and 52,752 square feet of heating surface.

There are two main condensers, each with 10,375 square feet of cooling surface.

Provision is made for a complement of 856 officers and men.

Bennington.—*Boiler Explosion.*—On the morning of July 21st, while getting up steam, preparatory to proceeding to sea, the collapse of a furnace in one of the *Bennington's* boilers resulted in the death of one officer and sixty-one enlisted men, the injury, more or less seriously, of forty others, and much damage to the ship.

The lower furnace of boiler B (the port forward one) collapsed, causing this boiler to be driven aft against boiler D, liberating the steam contained in both boilers, wrecking the steam piping and, through breaking of sea valve connections, permitting entrance of water to hull.

The vessel began to settle with heavy list to starboard,

making it necessary to beach her, with the assistance of tugs, to prevent sinking in deep water.

The heavy loss of life was caused principally by scalds and burns, due to escaping steam and water. No one in the fire-rooms escaped.

The accident is supposed to have been due to the accidental closing of the valve or pipe leading from boiler B to its steam gauge and the failure of the safety valve to lift upon the formation of an excessive steam pressure. The steam gauge, being disconnected, registered no pressure, and the water tender in charge was evidently not aware of the fact that pressure had formed until a few moments before the explosion, when a slight leak in the boiler was detected.

An examination and test of the material of the collapsed furnace showed it to be of good quality.

The boilers of the *Bennington* are about fourteen years old, having been in the ship since the time of her building in 1891; they are four in number, of the low, cylindrical, straightway, fire-tube type, with three furnaces each.

Wyoming—*Accident to.*—On the morning of July 18th, while at sea, *en route* from Panama to San Francisco, the U. S. S. *Wyoming*, lost her starboard propeller, the shaft having fractured just abaft the strut bearing. An examination by divers showed evidence of an old crack.

Paducah—*Trial of.*—The standardization and four-hour official trials of the gunboat *Paducah* were made at Newport, R. I., on August 15th and 16th.

The vessel, which was required by the contract to maintain an average speed of 12 knots for four consecutive hours, realized 12.85 knots without difficulty or mishap.

A complete description of this vessel and her sister ship, the *Dubuque*, appears in the first part of this number of the JOURNAL.

AUSTRIA.

Kaiman—*Launch of.*—The first-class torpedo boat *Kaiman*, built for the Austro-Hungarian Government by Messrs. Yarrow & Co., was successfully launched from their yard at Poplar

on Friday, June 2d. She is the first of a special type, the authorities in Vienna having determined to completely reorganize their torpedo flotilla upon most modern lines. The *Kaiman* will be propelled by a single screw. The machinery will be similar to that adopted in the Austrian destroyer *Huszár*, launched on the 31st ultimo by the same constructors, the difference between the two types of vessel being that the destroyer will be fitted with two sets of machinery, while in the torpedo boat there will be one set, the machinery in the two classes being identical, the object of which is to secure interchangeability throughout the torpedo flotilla to the fullest possible extent. A large number of these vessels is being constructed in Austria, the *Huszár* and *Kaiman*, built by Messrs. Yarrow, serving as patterns. The design of both types will be generally similar to corresponding vessels built by the same firm for the Japanese Navy, subject to improvements which recent experience has indicated as desirable.

The Destroyer Huszár.—Last week a trial was made, off the mouth of the Thames, of the destroyer *Huszár*, built by Messrs. Yarrow & Co. for the Austro-Hungarian Government. This vessel, as already stated in a former issue, has been built to serve as a pattern vessel for a large number of sister craft which are to be constructed in Austria, the naval authorities of that country having determined to reconstruct their torpedo flotilla. To this end they have also ordered from Messrs. Yarrow & Co. a first-class torpedo boat to serve as a model for similar boats which are to be built in Austria. Of these craft, five destroyers are being constructed at the Stabilimento Technico, of Trieste, and six will be built at Fiume. Nineteen of the torpedo boats will be built at the former establishment and ten at Fiume. The *Kaiman*, which is the name of the model torpedo boat launched at Poplar on June 2d, is 180 feet 6 inches long and 18 feet 6 inches wide, the draught being 8 feet 6 inches and the displacement 200 tons. The propelling engines will develop 3,000 indicated horsepower, and the speed is to be 25¾ knots with a load of 55 tons. The armament will consist of four 1.85-inch guns and three 18-inch

torpedo tubes. The vessel's complement will be two officers and twenty-five men.

To return to the destroyer tried last week, her length is 220 feet and her breadth 20 feet 6 inches, the depth being 13 feet. The total displacement with 100 tons on board—the contract trial weight—is 400 tons. The hull is constructed of the high-tensile steel which is made especially for torpedo craft. The conning tower is of chrome-steel and is bullet proof at a range of 20 yards point blank. The armament consists of two 18-inch torpedo tubes on deck, one 76-millimeter gun on a platform forward, and seven 47-millimeter quick-firing guns. Provision is made for carrying two spare torpedoes.

The propelling machinery consists of two sets of four-cylinder three-stage compound engines, driving twin three-bladed screws, each set developing 3,000 indicated horsepower, or 6,000 indicated horsepower collectively. The four-crank engine has been preferred, although it is heavier and more costly, because it gives less vibration than the three-cylinder arrangement. The engines are, of course, balanced on the Yarrow-Schlick-Tweedy system, and on the trial the *Huszár* ran at full speed in a most satisfactory manner in regard to vibration. Vibration, it may be added, is not only a question of convenience and comfort, but it has been found to be a matter that affects the health of the crew after a period of continuous running.

There are the usual auxiliaries, including electric-light installation for lighting the vessel and for the projectors, steam-steering gear and evaporating and distilling apparatus. There is a steam capstan forward. There are two 8-cwt. anchors of the ordinary Admiralty pattern. In order to provide for the housing of these, without masking the fire of the forward guns, a special tilting table has been devised by the Austrian authorities. The boilers are four in number, and are of the Yarrow straight-tube variety. The destroyer will have a complement of five officers and sixty-four men.

The *Huszár* is interesting, as she is similar—excepting in some details suggested by recent experience—to the destroyers

which Messrs. Yarrow & Co. built some short time ago for the Japanese Government, and which have been doing such excellent service for their side during what may, perhaps, be described as the late naval war. The Japanese boats, however, made a speed of 31 knots on trial, whilst the contract speed of the *Huszár* is 28 knots; but the Japanese boats were tried with a load of 40 tons, whilst, as stated, the trial load of the Austrian vessels is to be 100 tons. This is a reduction of speed of 1 knot for about every twenty tons additional weight.

The most marked difference between the Austrian and Japanese boats consists in the arrangement of the funnels and uptakes of the boilers. With these high-powered vessels the boiler space necessarily occupies a considerable length of the forward middle part of the hull, and this brings the foremost chimney very near the bow. This in turn pushes forward the navigating bridge, so that there is quite a short length of the vessel in front of the steering wheel. Every sailor knows how much easier it is to maneuver a vessel when the whole length is in front of him, but this cannot be arranged with modern steamers of this type, and, practically, the bridge must be in front of the funnels. With three-quarters or more of the ship out of sight, and that three-quarters swinging through the largest arc when the helm is put over, it is extremely difficult to maneuver these light craft in narrow or crowded waters, especially when a strong wind is blowing, whilst the tactical advantage of accurate steering can hardly be over estimated.

To get the boilers into the vessel and to keep the bridge a reasonable distance from the bow was the problem to be solved, and this has been done in the *Huszár* in a very simple manner. The chimney of the forward boiler—there is a funnel to each boiler—is carried aft by bringing the uptake horizontally over the stokehold by means of a raised superstructure, and in this way an addition of 17 feet or 18 feet of "sighting" is secured. There appears to be no serious obstacle to the plan being still further extended, so that the wheel could be placed still further aft, or the bridge being kept further away from the chimneys—a desirable feature with a leading wind.

The trial of last Thursday week was of a preliminary nature in regard to speed, the time being chiefly devoted to the usual anchor trials. Four runs were, however, made on the Maplin mile, the vessel realizing the contract speed of 28 knots with the boilers steaming easily, the pressure averaging about 200 pounds to the square inch, or 65 pounds less than that at which they are designed to work, and the revolutions averaging 380 per minute.

The official full-speed trials took place on Tuesday, June 20, when a speed of 28.537 knots was attained during a continuous run of three hours, the vessel being loaded with 100 tons. The Austro-Hungarian Government was represented by Captain Josef Ritter von Schwartz, naval attaché; Captain Franz Ritter von Keil; Baron Franz von Preuschen; Marine Engineer Rudolf Zhernotta; Herr Josef Grond; Engineer Anton Tonsa; Herr Hans Musurka, and Herr J. Seifridsberger. The vessels were constructed under the two last-named officials. Mr. W. W. Marriner and Mr. Harold Yarrow were in charge of the trials on behalf of Messrs. Yarrow & Co.—"Engineering," London, June 23.

BRAZIL.

The additions which Brazil proposes to make to her fleet are expected to comprise three iron clads, with a displacement of 12,500 tons to 13,000 tons each; three armor-plated cruisers, with a displacement of 9,200 tons to 9,500 tons each; six torpedo-boat destroyers, with a displacement of 400 tons each; six torpedo boats with a displacement of 50 tons each; three Holland submarines, and a transport with a displacement of 6,000 tons. It is proposed that the iron clads shall each carry sixteen guns; the armor-plated cruisers are to steam at the rate of 21 knots, and the torpedo-boat destroyers are to attain a speed of 31 knots. The execution of the program is to extend over nine years, divided into three terms of three years each. The cost which it will involve is estimated at £7,120,000.—"Engineering," London.

ENGLAND.

Hibernia—*Launch of.*—The battleship *Hibernia*, of the *King Edward VII* class, was launched at the Devonport dockyard on June 17th. The principal dimensions are: Length between perpendiculars, 425 feet; extreme breadth, 78 feet; draught of water forward, 26 feet 3 inches; aft, 27 feet 3 inches, and load displacement, 16,350 tons. The engines will consist of two independent sets of the vertical triple-expansion type, each having four cylinders. Each set will be capable of developing 9,000 indicated horsepower, with 120 revolutions per minute. The speed contracted for at full power is 18½ knots. There will be eighteen water-tube boilers of the Babcock & Wilcox type, and three cylindrical boilers. The armament consists of four 12-inch breech-loading guns, worked in two barbettes, four 9.2-inch breech-loading guns in four barbettes, ten 6-inch breech-loading guns worked in a battery, twelve 12-pounder, 18-cwt., quick-firing guns, two 12-pounder, 8-cwt., quick-firing guns, fourteen 3-pounder quick-firing guns, two .303 Maxim guns and four submerged torpedo tubes. The total complement of officers and men will be 781.

(For further data as to this type, see Vol. XVI, pp. 998 and 1315, and Vol. XVII, p. 134.)

Achilles.—This British first-class armored cruiser was launched on June 17 from the yard of Armstrong, Whitworth & Co., at Elswick. The *Achilles* is one of four vessels provided for in the naval estimates of 1903–4. These vessels are known as the Warrior class, built on French lines, with guns in single turrets placed high up. The *Achilles* is 480 feet long, 73 feet 6 inches broad, 40 feet 1½ inches deep, with a draught of 27 feet. Her displacement is 13,660 tons, and her armament is as follows: Six 9.2 and four 7.5 breechloading guns, two 12-pounders and twenty-four 3-pounder quick-firing guns, two Maxim guns and three submerged torpedo tubes. The armor belt is six inches thick. Her speed will be 22 knots an hour.

(For further details see Vol. XVII, p. 546.)

Roxburgh—*Trial of.*—This cruiser, built by the London and Glasgow Shipbuilding Co., Glasgow, for the Admiralty, pro-

ceeded to sea on her official trial trip recently. She is the fourth of the cruisers of the *Devonshire* class which have passed through their steam and gun trials, and her speed record, as the mean of four runs over the measured mile on the Clyde, was 23.63 knots (the highest attained by any vessel of the class), to attain which the *Roxburgh* developed more power than the others. The engines throughout the eight-hours' trial maintained an average of 21,857 indicated horsepower, which is 857 indicated horsepower over that guaranteed in the contract. The starboard engine made 143 revolutions per minute, and developed 10,734 indicated horsepower, whilst the port engine for 144 revolutions indicated 11,123 indicated horsepower. The air pressure in connection with the tank boilers was 1⅜ inches, and in the water-tube boilers 1 inch, the latter of the Durr type. The steam pressure was 204 pounds at the boilers, and the coal consumption worked out to 2.31 pounds per indicated horsepower per hour. In the first of the coal-consumption trials of thirty hours' duration, to Portsmouth, with the engines making 86½ revolutions per minute, the power averaged 4,634 indicated horsepower, giving the ship a speed of 14.4 knots. The coal consumption on this trial was at the rate of 2.09 pounds per indicated horsepower per hour. The gun trials having been carried out in the English Channel, the *Roxburgh* returned to the Clyde, making the second of the coal-consumption trials of thirty hours' duration, when, at an average rate of 127.4 revolutions, 15,005 indicated horsepower was shown, and the coal consumption 1.991 pounds per indicated horsepower per hour. On four runs over the measured mile at Skelmorlie, the mean speed was 21.54 knots.

(For detailed description of this class see Vol. XVII, pp. 430 and 443.)

Scout Skirmisher—*Trials of.*—H. M. S. *Skirmisher*, the second of the vessels of the class designed and constructed complete for commission by Messrs. Vickers Sons & Maxim, Limited, at the Naval Construction Works, Barrow-in-Furness, completed this week her specified trials, which, as in the case

of the other vessels, included a 96-hours' coal-consumption test, an 8-hours' full-power trial, as well as gun, circling and general maneuvering performances. The first of the vessels was the *Sentinel* (see JOURNAL A. S. N. E., Vol. XVII, page 208). She has now been in commission for some time, is now engaged on an extended cruise, and, peculiarly enough, was at anchor at the Tail of the Bank on the Clyde, to greet her sister ship on her return from the full-power test on June 6th. Although we have already described the vessels of the class, it may be said here that the Vickers ships are 360 feet in length, 40 feet in beam, and at their full load displacement of 2,940 tons they carry a coal supply sufficient to enable them to steam at a cruising speed of between 10 and 12 knots for 1,500 miles. The draught under those conditions is 14 feet 3 inches. The constructors guarantee a speed at load draught of 25 knots, and, to attain this, triple-expansion engines, driving twin screws, were fitted to develop 17,000 indicated horsepower. The *Sentinel* on her trials easily exceeded this guarantee, and now the *Skirmisher* has repeated the same satisfactory performance, her mean speed on the 8-hours' run being 25.2 knots, when developing 16,899 indicated horsepower.

The trials of the *Skirmisher* have been carried out within a comparatively short period of time. She only left Barrow on May 23, and after a preliminary trial in Morecambe Bay proceeded to the Clyde, where she arrived on May 24th. The first preliminary progressive trial was carried out on the 25th, when the vessel worked up to a speed of 20 knots. Two days later she continued the progressive runs, the machinery developing the necessary power to give the speed of over 25 knots. The vessel then proceeded on her 96-hours' coal-consumption trial, beginning on May 30 and terminating on June 3d. The performance on this trial showed the machinery to be economical. On the first half of the trial the speed attained was 10.6 knots for 904 indicated horsepower, and the coal consumption showed that 10.3 nautical miles were covered for each ton of coal consumed. On the second half of the trial

the results were equally satisfactory; the speed was 10.9 knots for 995 indicated horsepower, and the coal consumption was equal to 1 ton for each 10 nautical miles steamed. Table I gives the official results for the 96-hours' trial. With a mean steam pressure in the boilers of 177 pounds and at the high-pressure receiver of the engines of 44 pounds, the power for 79.6 revolutions was 949 indicated horsepower. This gave a speed of 10¾ knots, while the coal consumption worked out to practically one ton per 10 knots steamed. The coal consumption is debited with the cost of making up the loss of water in the evaporators; although only 15.9 tons were lost during the four days and nights of steaming, the evaporators made 16.67 tons. One more point which ought to be noticed is that the vessel had not been in drydock since April 17; and when later she was docked, preparatory to the 8-hours' full-power trial, it was found that the hull was by no means clean.

Table I.—RESULTS OF 96-HOURS' COAL-CONSUMPTION TRIALS OF H. M. S. *SKIRMISHER*, MAY 30 TO JUNE 3, AT CRUISING SPEED.

——	Starboard.	Port.
Mean steam in boilers, pounds	177	177
at engines (high-pressure receiver), pounds	47	41
air pressure in stokeholds	nil.	nil.
cut off in high-pressure cylinder, per cent	35	35
vacuum, inches	25.7	25.5
revolutions per minute	79.6	79.7
Mean pressure in cylinders:		
High, pounds	20.9	18.9
Intermediate, pounds	5.62	4.78
Low, forward, pounds	1.7	2.32
Low, aft, pounds	1.63	2.35
Indicated horsepower:		
High	210	189
Intermediate	144	123
Low, forward	60	83
Low, aft	57	83
Total indicated horsepower	471	478
Gross total indicated horsepower	949	
Coal, per indicated horsepower, pounds	2.5	
Sea miles per ton of coal	10.075	
Speed, knots	10.756	
Loss of water (tons per 1,000 indicated horsepower per 24 hours)	4.21	
Total loss of water for all 96 hours, tons	15.9	
Water made by evaporators, tons	16.67	

The eight-hours' full-speed trial was run on June 6th, and the weather conditions were satisfactory, there being only a slight breeze from the northeast. The vessel, soon after leaving the anchorage, commenced her trial at 9 o'clock, and six runs were made over the measured mile, occupying 1 hour and 25 minutes. The vessel thereupon continued down the Firth, and the speed for the whole eight hours was measured by the average number of revolutions required on the six runs over the measured mile for 25 knots. The revolutions for 25 knots are almost exactly 200, and throughout the succeeding 6½-hours' run the engines made 201.55 revolutions, the power being on the 6½-hours' run 17,013 indicated horsepower, while for the whole of the eight hours the mean was 16,899 indicated horsepower. The average speed was 25.2 knots. We append the mean results attained during the 6½ hours following the measured-mile performance:

Table II.—RESULTS OF FULL-POWER TRIAL OF H. M. S. *SKIRMISHER*, JUNE 6, 1905.

——	Starboard.	Port.
Mean steam in boilers, pounds	253	253
at engines (high-pressure receiver), pounds	239	239
Air pressure in stokeholds, inches	2.3	2.3
cut off in high-pressure cylinder, per cent	75	75
vacuum, inches	25.5	25.3
revolutions per minute	201.5	201.6
Mean pressure in cylinders:		
High, pounds	103.5	106.1
Intermediate, pounds	36.7	38.8
Low, forward, pounds	19.7	16.4
Low, aft, pounds	19	19.6
Indicated horsepower:		
High	2,624	2,693
Intermediate	2,382	2,520
Low, forward	1,757	1,465
Low, aft	1,818	1,754
Total indicated horsepower	8,581	8,432
Gross total indicated horsepower	17,013	
Water loss per 1,000 indicated horsepower per 24 hours, tons	2.33	
Total loss of water for all 8 hours, tons	13.16	

These results are very satisfactory. The starboard and port engines ran at almost the same speed, and gave practically the

same indicated horsepower. The trial concluded at 5 o'clock, and the ship anchored. On the completion of the gun, anchoring and circle trials on Thursday, the vessel returned to Barrow to be completed for commission. Messrs. Vickers Sons & Maxim, Limited, will be the first firm to deliver their pair of scouts.—"Engineering," London.

Scouts Pathfinder and Patrol—*Trials of.*—The length between perpendiculars is 370 feet; breadth, 38½ feet; and the vessels have a displacement at normal draught of 2,850 tons. The hull is built generally of mild steel, with special steel in parts of the structure subjected to the greatest stresses, and is well subdivided into numerous watertight compartments. A steel protective deck worked in the region of the water line (except over engines, where it is at the level of the upper deck,) extends from stem to stern, being ⅝-inch thick on the flat portions and 1½ inches thick on the slopes. Side armor of 2-inch K. N. C. steel is worked in the wake of the engines, extending from well below the water line to the height of the upper deck—an arrangement which conduces to large and airy engine rooms.

The *Pathfinder* encountered very rough weather on her earlier official trials, and won high praise for her steadiness in a seaway. The coal stowage in these two vessels was made a special feature of the design, and bunkers are provided both above and below the protective deck, with a capacity sufficient to give the vessel a radius of action of 6,000 miles at an economical speed.

The vessels all carry an armament of ten 12-pounder quick-firing guns of the new 18-cwt. pattern, eight 3-pounder quick-firing guns, and two 18 inch torpedo-tubes. Three of the 12-pounders are placed on the forecastle, three on the upper deck aft, and the remainder in the waist of the vessel. Cabin, messing and sleeping accommodation is provided for a complement of 286 officers and men.

The machinery consists of two sets of four-cylinder triple-expansion engines in separate watertight compartments, balanced on the Yarrow-Schlick-Tweedy system, the low-pres-

sure cylinders being placed at the ends. To obtain the full advantage of the keelsons for supporting the machinery, and to maintain the longitudinal keelsons as deep as possible, the engine-bed plates are made of special design, sunk in and secured between the longitudinals, thus securing a very strong and stiff arrangement with a minimum of weight. The valve gear is of the ordinary double-eccentric double-bar link type. Piston valves, taking steam inside, are fitted to the high and intermediate engines, the low-pressure being fitted with double-ported flat slide valves having relief rings at the back. The cylinders are of the following dimensions: High-pressure, 32½ inches; intermediate pressure, 51½ inches; and each low pressure, 58 inches in diameter, with a stroke of 2 feet 6 inches. The steel shafting is hollow throughout, the crank shaft being 13½ inches external diameter, with 6⅜ inches bore. The condenser shells are of sheet brass, 6 feet 2 inches in diameter, one for each engine, the total cooling surface being 14,000 square feet. Special arrangements have been made to ensure a minimum of vibration to the tubes. The air pumps are Weir's independent type, one double pump being fitted to each engine. The pipes to the centrifugal circulating pumps are 17 inches in diameter. Special attention has been given in the design to the lubricating arrangements on the main engines, the outcome of which has been satisfactory. There are twelve water-tube boilers of the well-known Laird type, fired from four stokeholes. The working pressure is 275 pounds per square inch. The total heating surface is 43,800 square feet, and the total grate area 656 square feet.

The contract trials for each boat consisted of a 96-hours' trial at between 10 and 12 knots and a full-power trial of eight hours, during which a speed of 25 knots had to be obtained. Throughout these trials the machinery worked in the most satisfactory manner, and in both vessels the trials were completed within a very short period. H. M. S. *Pathfinder* commenced her 96-hours' trial on March 26, and ran her full power on April 8. H. M. S. *Patrol* ran her 96-hours'

on June 1 to 5, and her full-power trial on June 17. The following are some of the results, the full-power records being those for 6½ hours after completing the miles.

	H. M. S. *Pathfinder.*		H. M. S. *Patrol.*	
	96 hours.	Full power.	96 hours.	Full power.
Speed (knots)	10.988	25.345	10.969	25.062
Revolutions	85.4	220.2	84.7	213.5
I.H.P.	1,063	17,235	1,164	16,433
Coal per I.H.P., pounds	2.35	...	2.18	...
Water loss per 1,000 I.H.P., per 24 hours, tons	1.82	3.13	1.71	1.33

All steering and maneuvering trials were successfully carried out, and both vessels are rapidly approaching completion for commission; in fact, the *Pathfinder* will leave Birkenhead during the first week of July, well within her contract date.—"Engineering," London.

Steam Trials of H. M. S. Foresight—The scout *Foresight*—the second of the vessels of this type, built by the Fairfield Shipbuilding and Engineering Company—has concluded her official trials with most satisfactory results. This vessel is in all respects the same as the *Forward*, which completed her trials in May last; and as we then entered into details of the design of the ship and her machinery, we need not refer further to this matter than to say that the length of the vessel between perpendiculars is 365 feet and the beam 38 feet 9 inches. The designed displacement is 2,945 tons when the vessel has a draught of 14 feet 5 inches. The machinery is of the four-cylinder triple-expansion type, designed to develop 16,500 horsepower; but, as will presently be shown, the *Foresight*, like her sister ship, the *Forward*, exceeded the required speed, the power being only 14,300 horsepower, while in the *Forward* it was 15,000 horsepower. The boilers are of the modified Thornycroft type, with 42,960 square feet of tube surface and 752.4 square feet of grate area.

The first trial consisted in a coal-consumption test of 98 hours' duration at an economical cruising speed, and, as in

the case of the *Forward*, 10½ knots was found to be the most satisfactory rate. This speed, as is shown in Table I, was realized for 810 horsepower; the rate of coal consumption being 2.5 pounds, which is an improvement on the *Forward*. In that vessel the power was 29 indicated horsepower greater and the coal consumption 0.16 pound per unit higher.

Table I.—RESULTS OF 96-HOURS' COAL-CONSUMPTION TRIAL OF H. M. S. *FORESIGHT*, JUNE 24, 25, 26 AND 27, 1905, AT 10.552 KNOTS SPEED.

Draught, forward, 13 feet, 2½ inches; aft, 14 feet, 10½ inches.

	Star-board.	Port.
Mean steam in boilers, pounds	167	167
at engines (high-pressure receiver), pounds	50	53
air pressure in stokeholds	nil.	nil.
cut-off in high-pressure cylinder, per cent	33	33
vacuum, inches	25	23.8
revolutions per minute	80.47	80.16
Pressure in cylinders:		
High, pounds	15.75	17.4
Intermediate, pounds	4.55	5.15
Low, forward, pounds	2.17	1.85
aft, pounds	2.17	1.9
Indicated horsepower:		
High	151	166
Intermediate	111	125
Low, forward	70	58
aft	70	59
Total indicated horsepower	402	408
Gross total indicated horsepower	810	
Coal, per indicated horsepower, per hour, pounds	2.5	
Water loss per 1,000 indicated horsepower, per 24 hours, tons	3.56	
Indicated horsepower on mile	829	
Speed, mean of 24 runs on mile, knots	10.697	
96 hours, knots	10.552	

As a consequence of the reduced consumption the coal to be carried on the full-power trial was considerably less, so that the displacement was much below that designed for. Loaded with coal to enable her to run at 10½ knots for 1,500 miles at the rate of coal consumption and the power mentioned, the vessel proceeded on June 30 on her full-power test, which was to consist of six runs over the measured mile at Skelmorlie, steaming for 6½ hours afterwards, making 8 hours in all, at a guaranteed rate of 25 knots. This result was easily realized, the air pressure in the stokeholds throughout the trials being between 1 inch to 1½ inches, while the cut-off

at the high-pressure engines was at 60 per cent. of the stroke. As shown in Table II, the engines worked at practically 211 revolutions, and averaged 14,330 indicated horsepower, the mean speed being 25.224 knots.

Table II.—H. M. S. *FORESIGHT.* SIX RUNS ON MEASURED MILE AT FULL SPEED, JUNE 30, 1905.

Run.	Revolutions.	Speed.	Indicated horse-power.
		knots.	
1	206.7	25.175	13,723
2	214	25.14	14,956
3	211.2	25.46	14,732
4	211.2	25.035	14,770
5	208 7	25.316	14,029
6	207.5	24.76	13,772
Mean	210.99	25.224	14,330

This is slightly better than in the case of the *Forward*, which for 15,000 horsepower gave 25.286 knots. The difference, however, is probably accounted for by the less amount

Table III.—RESULTS OF FULL-POWER EIGHT-HOURS' TRIAL OF H. M. S. *FORESIGHT*, JUNE 30, 1905.

——	Mean of six runs on measured mile.		Mean of subsequent 6½ hours of trial.	
	Starboard.	*Port.*	*Starboard.*	*Port.*
Mean steam in boilers, pounds.....	245	245	245	245
Mean steam at engines (high-pressure receiver), pounds.....	230.5	229	232.8	232
Mean air-pressure in stokeholds, inches..	1	1	1.4	1.4
Mean cut-off in high-pressure cylinder, per cent.....	60	60	60	60
Mean vacuum, inches.....	25	25	25	24.7
Mean revolutions per minute.....	210.1	209.6	210.3	210
Mean pressure in cylinders:				
High, pounds.....	107.6	110	107.4	110
Intermediate, pounds.....	33.4	33.4	32.6	32.5
Low, forward, pounds.....	14.4	14	14.3	14.5
aft, pounds.....	14.3	14.6	14.5	15
Indicated horsepower, high.....	2,673	2,734	2,669	2,731
intermediate	2,131	2,129	2,114	2,069
low, forward.....	1,175	1,136	1,167	1,169
aft.....	1,161	1,191	1,179	1,210
Total indicated horsepower.....	7,140	7,190	7,129	7,179
Gross total indicated horsepower.....	14,330		14,308	
Total loss of water for all 8 hours, tons..	5.17			
Water loss per 1,000 I.H.P. per 24 hours, tons.....	1.08			
Speed, knots.....	25.224		25.123	

of coal required to be carried in the bunkers, so that when the bunkers are carrying their full load the vessel will easily exceed her speed while giving a much greater radius of action. In Table III the result of the engine's performance, not only on the six runs over the measured mile, but on the subsequent 6½-hours' trial, are set out.

Throughout the whole six hours the vessel steamed very easily, while the loss of water was only about 1 ton per 1,000 horsepower per 24 hours, as compared with 2.71 tons in the case of the *Forward.*—"Engineering," London.

Dreadnought.—The new British battleship *Dreadnought*, included in the navy program of the current financial year, is to excel any vessel yet built, and for this purpose will be fitted with ten 12-inch guns. Hitherto no ship has had more than four of these large weapons, and although this latest vessel will not carry any other pieces except the machine guns for warding off torpedo attack, her fighting power will be far in excess of that of her predecessors. Prior to the vessels of the *King Edward* class the usual armament for British battleships was four 12-inch guns and ten or twelve 6-inch quick-firers, the collective muzzle energy of the sixteen guns being about 210,000 foot tons. In the *King Edward* class an improvement was made by adopting four 9.2-inch guns in addition to four 12-inch and ten 6-inch weapons. This increased the muzzle energy of the guns collectively to about 275,000 foot tons. In the *Lord Nelson* class the 6-inch weapons were eliminated and four 12-inch and ten 9.2-inch adopted. At the same time it was decided to increase the length and power of each weapon, so that the total muzzle energy from one round from all the guns became 433,000 foot tons. Now the 9.2-inch gun is discarded and only 12-inch weapons installed, and the ten of these together will give an energy of 480,000 foot tons. This change, it is pointed out, simplifies the arrangements for ammunition on board and at naval bases, so that after action a ship can easily get a new supply. The battleships will be of 18,000 tons displacement, and fitted with turbine machinery and Babcock & Wilcox boilers of 23,000

indicated horsepower, giving a speed of about 21 knots, as compared with 19 knots for the ships of the *King Edward* class.—"Army and Navy Register."

Destroyers.—Contracts have now been arranged for the construction of five sea-going destroyers included in the navy program for the current year, and these vessels will be built, one each by Messrs. J. I. Thornycroft, Limited, Chiswick; Sir W. G. Armstrong, Whitworth & Co., Limited, Elswick; Messrs. Cammell, Laird & Co., Limited, Birkenhead; Messrs. R. and W. Hawthorn, Leslie & Co., Limited, Newcastle-on-Tyne; and Messrs. J. S. White & Co., Cowes. These vessels are a great advance upon anything that has yet been done in torpedo-boat destroyers, as the speed guaranteed, under penalty, by the contracting firms is to be 33 knots when the vessels are carrying a load of coal sufficient to give them a radius of action of 3,000 miles at about 10 knots speed. This condition is much more severe than was exacted in the case of the 30-knot torpedo boats; and, moreover, the scantlings proposed for these high-speed craft will be considerably heavier. It is true that speeds of 32 to 33 knots were attempted by one or two builders in connection with the 1897–8 program, but the highest realized on trial was 31½ knots. Even then the boats were not required to meet the same stringent conditions as is the case with these new 33-knot craft. The boilers are to be of the Express type, adaptable for using oil fuel, and the propelling machinery in all cases will be of the Parsons turbine type, with arrangements similar to those in the *Amethyst*, where special turbines were fitted for low-speed cruising. The displacements of the boats will average over 800 tons, so that very considerable power will require to be developed to attain the speed. It may be noted that the 30-knot destroyers were from 300 to 360 tons displacement. When the scantlings were increased for what is now known as the "River" class, the displacement was increased from 525 to 550 tons; but the speed was reduced to 25½ knots, under the same fuel-capacity conditions as in the new boats. It will therefore be seen that the new vessels will mark a great step in speed, and

the contracts carry with them corresponding responsibilities. It is the intention to order later an experimental destroyer, which it is expected will have a speed of 36 knots.—"Engineering," London.

Submarine A8—*Accident to.*—Another deplorable disaster to a submarine occurred at Plymouth on June 8th, when *A8*, while at exercise for instructional purposes off the breakwater, foundered without giving any warning to her unfortunate crew. The conning towers were open, when suddenly the bow of the vessel dipped heavily, which caused her to ship a large quantity of water through the conning tower, and before it was possible to close the conning tower she went down. Lieutenant Candy (the commander), Sub-Lieutenant Murdock, and two petty officers, who were on deck at the time, were saved; but Sub-Lieutenant Fletcher and nine of the crew, who were below, were all drowned. Steps were immediately taken to raise the vessel, which was successfully accomplished on the morning of June 12th. The finding of the courtmartial which was held on the survivors was as follows:

"The Court finds that submarine *A8* was lost outside Plymouth breakwater, about 10.30 A. M., on the 8th June, through foundering from water getting in through the open conning tower when running at ten knots speed with a buoyancy at starting of six tons and trimmed four degrees by the stern, which buoyancy and trim appeared to have altered from some unexplained cause, such as the movement of weights forward, possibly admission of water, or free water. There is no evidence to show which way the diving rudders were put, although repeated orders were given to put them up. They may, however, have been put inadvertently the opposite way to that actually intended. Water having once commenced to get into the conning tower and running forward, she could not recover herself, although the helm was finally found hard up.

"The Court regrets that the officer in command did not stop the engines sooner when he realized that his repeated orders

to put the diving rudders up had no effect, although expert evidence shows there was no reason to anticipate danger in running the boat trimmed as at starting, as it appears to have been considered impossible for her to dive under these conditions. The Court, therefore, acquits Lieutenant Algernon Hugh Chester Candy and the other survivors of submarine *A8* of all blame for the loss of that vessel."

The accident to *A8* is the third to British submarines which has been attended with fatal results. In March of last year *A1* was run down at Spithead, with a loss of eleven lives. In February of this year there was an explosion of petrol vapor on board *A5* at Queenstown, by which four men met their death.—"J. R. U. S. I."

Magnificent—*Accident on Board.*—A serious accident occurred on board the *Magnificent* on June 14th, when at target practice off Tetuan, on the Moorish coast, by which an officer and four men lost their lives and twelve others were more or less injured. It appears there was a miss-fire in one of the 6-inch guns; the breech of the gun was opened to ascertain the cause, when the cartridge exploded, igniting two others in the casemate.

FRANCE.

Submarines.—In view of the fact, as announced in these columns recently, that the French naval authorities have recently launched one of the largest submarine boats in the world, her displacement being 213 tons, and her engines having 250 horsepower, it is interesting to observe that the French are also evolving submarines of so small a type that they can be carried on battleships and cruisers, to be placed in the water only when needed. One boat of this type, the *Volta*, now under construction, is said to be only 36 feet long. The maximum diameter of her circular cross sections, according to the "London Times," is 6 feet 9 inches, and her displacement is less than 20 tons. We learn from the same authority that she will use electrical power alone, and that her estimated speed limit is about eight miles an hour. She is to have a crew of three men only and to carry two torpedoes, one slung on each

side with suitable detaching gear. It is stated that the preliminary trials have been successful, and it is to be hoped that further information concerning this novel vessel may soon be forthcoming. This new type of submarine is being constructed by private builders under government supervision, and the experiment is a matter of keen interest to submarine experts in Europe. This is the first time a private company has undertaken to build a submarine in France, and the attitude of the French Admiralty is watched with great interest by all concerned, as they have not themselves undertaken the construction of similar classes of submarines, but have pursued a policy of continuous development in size and cost.—"Army and Navy Journal."

Destroyers.—Commenting on the prominence to be given to destroyers in the naval program of France, as fifteen of this type of vessel are to be put under construction in the French shipyards this year, a much larger number than has been laid down at one time in any previous year, the "London Engineer" says that the introduction of the scout has induced the French naval department to turn its attention to the necessity of efficiently armoring the new destroyers so as to protect the engines and boilers from light artillery at any range, while they will be invulnerable to the fire of guns of 76 mm. bore at distance from 2,000 m. to 3,000 m. The danger of a destroyer being put out of action or sunk by quick-firing guns has been so far increased by the superior armament and higher speed of the scout that the destroyer can no longer hope for safety in flight, and in the opinion apparently of M. Normand, who is responsible for the designs of the destroyers, the efficiency of these craft can only be maintained by protecting the vital parts against such fire as may be brought to bear upon them by the scout. In the new destroyers the displacement will be increased from 330 to 450 or, perhaps, 470 tons, so that the engines, boilers and other vital parts may be protected by armor plate 40 mm. to 50 mm. in thickness. The speed will be two knots less than that of the 330-ton craft, but they are expected

to be more seaworthy in heavy seas.—"Army and Navy Journal."

Farfadet—*Accident to.*—The French submarine *Farfadet* met with an accident, July 6, at the entrance to the port of Sidi Abdallah, Tunis, under the following circumstances: Commandant Ratier, at the moment of plunging, noticed that one of the doors was not closed properly and rushed to try to adjust it hermetically. He was too late, the water rushing in with force and throwing out the commandant and two men, the boat sinking in ten meters and lying in the mud at the bottom. Twelve men were encased in the boat, and powerful tugs and a strong contingent of engineers have been working for a week, but without success, to rescue the men and bring up the boat. The entombed men, when divers went down on July 7, were reported alive in the after part of the submarine, and rapped on the inside of the boat to communicate with the divers. On July 7 the *Farfadet* was raised above the surface of the water long enough to ascertain that the occupants were alive, and fresh air was pumped into the vessel; but when an attempt was made to tow her into shallow water she sank again. The boat was raised again on July 9, and the imprisoned men were reported still alive, but almost on the point of suffocation. When the hatch had been got about a yard out of the water the crane broke and the boat sank again. A second attempt failed to raise the boat, and it was later reported that the men in her no longer responded to signals. The *Farfadet*, which is 125 feet long, has electric propulsion, four torpedo tubes, and has reversible propellers. Her air tanks and cabin accommodations are exceptionally good for a submarine boat, and this is attributed as the reason the prisoners were able to keep alive for so long a time.—"Army and Navy Journal."

GERMANY.

New Ships.—An iron clad, officially known at present as *Q*, and a division of torpedo boats, officially known as *G132*, *G133*, *G134*, *G135*, *G136* and *G137*, have been ordered to be built in the Germania yards at Kiel. An iron clad, offi-

cially known at present as *R*, is to be built in the Schichau yards at Dantzic; an armor-plated cruiser, officially known as *D*, in the Blohm and Voss yards at Hamburg; and a small cruiser, officially known as *P*, in the navy yards at Dantzic.—"Engineering."

S124—*Accident to.*—The German battleship *Woerth* rammed torpedo-boat destroyer *S124* in the harbor of Eckernfoerde, July 5, cutting her completely in two, and three firemen in the boiler room, it is reported, were scalded to death by escaping steam. The *S124* was one of six torpedo-boat destroyers launched in 1904. She was of 420 tons displacement, and credited with a speed of 30 miles an hour.

River Gunboat.—A new river gunboat, which is to be sent out to Hong Kong in sections, has been constructed by the Schichau firm, Elbing. Her dimensions are as follows: Length, 157 feet 6 inches; beam, 26 feet 4 inches; draught, 2 feet, with a displacement of 168 tons. Her two engines will develop together 1,300 I.H.P., giving her a speed of 13 knots. Steam will be provided by two Thornycroft water-tube boilers. Her bunkers will hold 80 tons. She has a military mast, and is armed with one 3.46-inch 30-caliber quick-firing gun, one 1.97-inch 40-caliber quick-firing gun, and three 0.315-inch machine guns. Her complement will be fifty-three officers and men. Her deck house and sides are protected by a light armor of nickel-steel 0.315-inch in thickness.—"Neue Preussische Kreuz-Zeitung."

S125—*Trial of.*—The trial trip of the torpedo boat *S125*, built at the Schichau Works, Germany, and fitted with turbine engines, took place off Elbing, June 14. A dispatch states that she averaged 27.8 knots per hour for three hours. Her highest speed rate was 29½ knots per hour. Four sister boats, fitted with piston engines, averaged two knots per hour less than the *S125*.

JAPAN.

Katori—*Launch of.*—The battle ship *Katori* was successfully launched from the works of Messrs. Vickers Sons & Maxim on July 4th. A brief description of the vessel follows.

The length between perpendiculars is 420 feet, and over all 455 feet 9 inches; the breadth is 78 feet, and the depth to upper deck, 44 feet. The draught in fighting condition will be 27 feet, when the displacement will be 15,950 tons. As to gun fire—which is the most important element in a ship of the line—the *Katori* will mount pairs of 12-inch guns in barbettes at the forward and aft end of the upper deck. These are of the Vickers type, weighing 57 tons, and they will deliver their 850-pound projectiles with a velocity and energy which will enable them at six miles range to perforate armor 9 inches in thickness, and at four miles range to defeat modern armor about 13 inches in thickness. There are four 10-inch guns, mounted singly in barbettes at each corner of the main citadel. The Vickers guns of this type fire a 500-pound shot with an energy sufficient to perforate at six miles range armor 6½ inches thick, and at four miles range, 9 inches thick. In the armament of the British vessels of the *King Edward VII* class the guns corresponding to these 10-inch weapons are only of 9.2 inches caliber, so that the Japanese ship has a considerable superiority. In addition, the *Katori* mounts twelve 6-inch guns within a concentrated casemate amidships; each gun, however, is entirely separated by armor screens.

This completes the main armament of the *Katori*. One round from these twenty main guns represents a collective energy of nearly 375,000 foot tons. As the 12-inch guns may fire two projectiles per minute, the 10-inch guns three rounds, and each 6-inch gun ten rounds, the total weight of shot which may be discharged in a minute is 24,800 pounds, representing a collective energy of approximately 1,400,000 foot tons. All four 12-inch guns may be trained on the broadside. The two 10-inch guns and six 6-inch guns are similarly utilisable on the broadside, so that one minute's possible broadside fire is 15,800 pounds, with a collective energy of over 900,000 foot tons. As two 12-inch guns, two 10-inch guns and two 6-inch guns may be fired ahead, the vessel may maintain a very hot pursuit, discharging per minute four projectiles of 850 pounds, six of 500 pounds, and twenty of 100 pounds.

For the repelling of torpedo attack there are mounted, in protected positions in various parts of the ship, twelve 12½-pounder, three 3-pounder and six Maxim guns. Finally, the *Katori* has five submerged tubes for the firing of 18-inch torpedoes.

The armor on the new ship is arranged on the system first introduced into the *Mikasa*, ten of the 6-inch guns being in a concentrated casemate amidships instead of in a series of separate casemates. The whole of the broadside, from some distance below the water line to the upper deck, is armored. The main belt is 9 inches, reduced gradually to 4 inches towards the stem, and to 3½ inches at the stern. The minimum thickness of the upper parts of the broadside armor is 6 inches. Armored bulkheads are carried across the ship at the forward and aft ends of the citadel. Ten of the 6-inch guns are placed within this 6-inch armor, but each is isolated from its neighbors by transverse and longitudinal armor bulkheads. On the upper deck there has been constructed for the two remaining 6-inch guns a special armored battery. These guns have a very advantageous position for dealing with torpedo attack. The 12-inch gun barbettes have armor 10 inches in thickness generally, and those for the 10-inch guns have 6-inch armor. The ammunition hoists of all the guns are armored.

The third element in design has reference to the speed, and in this case the specified rate is 18½ knots; but as the *Mikasa* very considerably exceeded her designed rate, it would not be surprising if in this case also 19 knots is realized. The propelling machinery is of the type now usually adopted in battleships, there being two sets of four-cylinder triple-expansion engines, balanced on the Yarrow-Schlick-Tweedy system. The high-pressure cylinder is 35½ inches, the intermediate 56 inches, and the low-pressure cylinders each 63 inches in diameter, the stroke in all cases being 48 inches. It is anticipated that with 120 revolutions per minute and steam at 200 pounds pressure, the engines will develop 16,000 indicated horsepower. The steam pressure at the boilers is to be 230 pounds per square inch. The engines are designed to turn the pro-

pellers inwards when going ahead, and the starting platform is in the center of the ship. Wrought-steel columns form the supports of the cylinders, which are independent castings, while the back supports are of the ordinary cast-iron A-frame, with ample slipper-guide surfaces. This gives a fine, open engine. Stephenson link motion is adopted for working the valves, which are of the piston type on the high-pressure and intermediate cylinders, and of the double-ported flat design on the low-pressure cylinders. The crank shaft is in two interchangeable pieces, and the propeller shaft is 18 inches in diameter, with a 10-inch hole. The propellers have four blades, the diameter being 17 feet 3 inches; they are made entirely of bronze. There are four condensers, placed in the wings of the ship, the collective cooling surface being 17,000 square feet. As to the boilers, these are of the latest improved Niclausse type, twenty in number, located in three separate boiler rooms. The heating surface is 44,000 square feet—a ratio of 2.75 square feet per indicated horsepower. The total grate area is 1,334 square feet. There is, of course, a complete system of pumps, with the addition of independent air pumps, which are now almost universally adopted in connection with large high-power marine installations.

The total coal capacity is 2,100 tons, which will give the ship a high radius of action. Special attention has been given to the ventilation and general comfort of the living quarters of the ship. The thermo-tank system of ventilation has been fitted for the cabins and crew spaces, and for ventilating and cooling the magazines and shell rooms. A complete system of steam heating is adopted for the officers' quarters and crew spaces, and electricity is used in connection with ventilation as well as for lighting the ship. The vessel will have a total complement of 980 officers and men.

—"Engineering," London.

PERU.

The Peruvian Government has placed an order with Messrs. Vickers, Sons & Maxim for a cruiser of 24 knots speed closely

resembling, but slightly larger than, the scouts *Sentinel* and *Skirmisher*, which were built by the company for the British Navy.

RUSSIA.

New Ships.—The Russian battleship *Andrei Pervozvannyi*, of 16,630 tons, was launched at the Galerny Island yard, St. Petersburg, recently, and a sister ship, the *Pavel I*, is in hand in the Baltic yard. A great deal of uncertainty surrounds the new Russian ships, but a number are building from funds raised by private subscriptions. One of these, the torpedo cruiser *Dobrovoletz*, was launched at the Putiloff yard on June 11. The "Yacht" says that Russia is preparing to build, or actually building, three armored cruisers of the *Bayan* class—the *Admiral Makaroff*, *Pallada* and a new *Bayan;* as also four gunboats of the *Gyliak* class, named after the vessels lost—*Gyliak*, *Korietz*, *Sivoutch* and *Bobr*. Ten river gunboats are also mentioned—the *Oratschanin*, *Mongol*, *Buriat*, *Saboikaletz*, *Sibiriek*, *Koreia*, *Kirgis*, *Kalmuk*, *Amuretz* and *Ussurietz;* two destroyers of 300 tons, the *Lietnant Melejeff* and *Inzhener Mechanik Anastasoff*. In addition to these the names of twenty other destroyers are given, some of them being those of destroyers lost in the present war, and others named after officers who have distinguished themselves—Captain Jurassovky, Lieutenant Sergeieff, and Engineers Sverieff and Dmitrieff.—"Army and Navy Journal."

SWEDEN.

Magne—*Launch of.*—The torpedo-boat destroyer *Magne*, for the Royal Swedish Government, was launched on Wednesday, August 2d, from Messrs. Thornycroft's Church Wharf Works. She is a vessel of the *Shirakumo* type, which has proved so successful in the present Russo-Japanese War. Her guaranteed speed is 30½ knots, to be maintained for three hours with a load of 55 tons on board. The vessel is 219 feet 9 inches long over all, 20 feet 9 inches beam, and has a draught of 8 feet 3 inches at the screws. The machinery, which is over 7,200 indicated horsepower, consists of two sets of four-cylin-

der vertical triple-compound condensing engines and four Thornycroft water-tube boilers, worked under forced draft on the closed-stokehold principle. The bunkers have a capacity for 80 tons of coal, giving a radius of action of 2,800 nautical miles at a speed of 15 knots per hour. The armament consists of six 57-millimeter quick-firing guns—one mounted on the conning tower, one aft, and four on the broadside. There are also two 45-centimeter revolving torpedo-tubes. The boat is extensively sub-divided into watertight compartments by transverse bulkheads and watertight decks under the living spaces.—"Engineering," London.

MERCHANT SHIPS.

Viking.—The new turbine steamer *Viking*, built for the Isle of Man Steam Packet Company by Sir W. G. Armstrong, Whitworth & Co., completed her trial the latter part of June.

The principal dimensions are: Length between perpendiculars, 350 feet; length over all, 361 feet; breadth, extreme, 42 feet; depth to main deck, 17 feet 3 inches; depth to shelter deck, 25 feet 3 inches.

The subdivision of the vessel into watertight compartments has received special attention, and the principle has been carried so far that the ship will float with any two compartments full of water. The vessel has been built to Lloyd's requirements, and to conform with all the rules of the Board of Trade for a passenger certificate. To meet the special requirements of this trade she is designed to carry a very large number of passengers—nearly 2,000—and to afford them ample accommodation in the way of promenade space, together with extensive facilities for refreshment in the shape of numerous dining saloons and bars. Her shelter deck extends uninterruptedly from stem to stern, a length of 361 feet, and above it is a big promenade deck, about two-thirds this length, which is carried out to the ship's sides, and is thus 42 feet wide at its broadest point. The first-class accommodation is aft and the second-class forward, the middle part of the ship being mainly occupied by the boilers and engine room. The main, upper and shelter decks are continuous, except in way of the engine and boiler rooms.

Abaft the main entrance hall, on the shelter deck, there are four private cabins and a tea room, close to which there are two of the companion ladders leading to the promenade deck above.

The main saloon is the full width of the vessel, and is

lighted by skylight and large rectangular windows at the sides. The woodwork is in dark oak, and a series of black-and-white reproductions of famous pictures, by well-known artists, forms a pleasing contrast with the dark tones of the somewhat severe paneling, which is also broken by the deep-set window openings, and relieved by a simple carved design. The entrance hall is paneled in oak, of the same tone as the main saloon, combined with blue and white Dutch tiles.

The ladies' room is entered through folding doors on the port side of the main saloon. It is paneled in sycamore in simple style, and is lighted by large windows in the ship's side and by the large light and air trunk which leads to the dining saloon on the deck below. A large lavatory opens off the ladies' room.

At the after end of the main saloon a stairway leads down into another saloon, where accommodation is provided for those who wish to lie down.

The dining saloon is entered by means of a stairway alongside the main entrance to the saloon. It provides seating accommodation for about 100 people, and is decorated with oak paneling. The usual pantry arrangements, with hot plates, lift to the galley, &c., are provided, and all the accessories required to make the service in this department as expeditious as possible.

The first-class smoking room is on the shelter deck, and is entered by doors placed just forward of the stairway from the shelter to the upper deck.

The second-class accommodation is at the forward end of the vessel, and consists of two large general saloons, one of which is arranged for dining, and a ladies' saloon. Lavatory accommodation is provided in convenient places throughout the vessel.

The propelling machinery consists of three sets of turbines manufactured by the Parsons Marine Steam Turbine Company. The high-pressure turbine is in the center, with a low-pressure turbine on each side. The surface condensers are in the wings at the after end of the engine room, with the centrifugal

circulating pumps close alongside them. Two large independent air pumps, of Messrs. Weir's make, are fitted conveniently near the condenser, and discharge into a large feed tank, whence the feed water is led to the automatic float-control tank, and so to the two feed pumps, which are also of Messrs. Weir's make. These pumps discharge to the boilers direct, or through an exhaust feed heater on Parsons' patent, which gave very satisfactory results on the trials. The Parsons' patent vacuum augmenter is fitted in connection with the main condenser, enabling a high vacuum to be maintained. The lubrication of the bearings is effected by means of an oil-circulating pump, which drives the oil, under considerable pressure, through all the bearings, and into a cooling and filtering apparatus on its way back to the pump. The stern-way turbines are arranged as usual within the low-pressure casings, and revolve idly *in vacuo* while the vessel is steaming ahead. The three shafts are of Armstrong-Whitworth Openshaw steel, and the propellers of a special bronze mixture worked up to a highly-polished surface.

Steam of 160 pounds pressure is supplied by four double-ended boilers, 15 feet in diameter and 19 feet 6 inches long, placed in two boiler rooms separated by a watertight bulkhead. The stokeholds are arranged for forced draft, and the air is supplied by four fans placed in a separate compartment on the shelter deck, and communicating with the stokeholds by means of straight air trunks. An emergency exit from each boiler room is arranged through the air lock into the fan room. The funnels are fitted with outer casings which are utilized as upcast ventilators, and for this purpose are connected to compartments in various parts of the ship. A spark arrester, Messrs. Denny & Brace's patent, is fitted in each funnel. The result of its trial was most satisfactory, as there was no sign of sparks or cinders when running at the maximum speed. The whole of the stokehold and boiler-feeding arrangements were carried out by the Wallsend Slipway and Engineering Company.

Two sets of trials were carried out, irrespective of the

builder's preliminary test. The first official trial took place over the measured mile off the mouth of the Tyne, under somewhat unfavorable weather conditions. There was a fresh northeasterly breeze and from 2 feet to 3 feet of sea. The air pressure in the stokeholds did not exceed ¾-inch water column, and no attempt was made to develop the maximum power of the boilers. On June 14 the vessel was taken to sea for an extended trial. A course was set from Souter Point to Flamborough Head, 68.277 nautical miles, and the distance was traversed in both directions, care being taken to eliminate, as far as possible, the effect of the tide. The southward run was made at a speed of 23.684 knots, and the northward at 23.375 knots, or a mean speed over the whole distance of 23.53 knots. The vessel was kept at full speed for nearly seven hours. Although the power developed on this trial was higher than on the preliminary trials, there was very considerable reserve in the boilers.

The maneuvering qualities of the ship are very favorable.

Wm. E. Corey—*Launch of.*—The steamer *Wm. E. Corey*, building for the Pittsburg Steamship Co., was launched from the South Chicago yard of the American Shipbuilding Co. on June 24th.

The *Corey* is the second of the quartet of giant steamers building for the Pittsburg Steamship Co. to be launched. She differs from the others only in cabin arrangement. The other three ships have no passenger accommodations whatever, having only one spare room to accommodate an occasional representative of the company. The passenger accommodations of the *Corey*, however, are quite complete, the staterooms, five in number, being decorated in birch and enameled white, each stateroom having a private bath. There is also an observation room on the forecastle deck with unusually large windows. This room is decorated in Circassian walnut. The passenger accommodations, of course, include a private dining room. The *Corey* is, of course, the flagship of the fleet.

The *Corey* and her sisters offer further testimony to the originality of design now marking the evolution of shipbuild-

ing on the lakes. They are in a number of features unlike any ships built or building. In the first place, they are the longest on the lakes, being nine feet longer than the *Augustus B. Wolvin*, or 569 feet over all, 549 feet keel, 56 feet beam and 31 feet deep. While the general principle of construction is the same as their predecessors, in the absence of stanchions and stringers or any interference below deck with the unloading machines, the vessels are not of arch construction. Mr. Colby, after consulting a number of authorities, concluded that an arch was primarily intended to support weight from the top, and that a more rigid longitudinal construction could be secured by a straight girder across the deck.

The construction of the cargo hold also is a departure from anything at present on the lakes. The cargo hold descends in a straight line from the deck until within 9 feet of the tank top, when an inner hold or hopper is constructed 9 feet from the skin of the ship and descends to the tank top. The top of the hopper is joined to the skin of the ship by an incline construction, thus avoiding what would otherwise constitute a shelf and also preventing any lodgment of ore. The water bottom is 5 feet 6 inches deep, and is used for water ballast in conjunction with a 9-foot space formed by the hopper and the sides of the ship. This construction gives enormous water-ballast capacity, approximating 8,500 tons, making the steamer navigable in all sorts of weather and providing extraordinary rigidity in the longitudinal section. The hatches number thirty-four, and are spaced 12-feet centers, the hatch covers being operated by the Brosseau device, enabling the hatches to be opened or closed from a central station. These ships are designed for a carrying capacity of 10,000 tons, but when favored with deep draught their actual cargoes are likely to be considerably in excess of this figure, as has already been proved by the initial trip of the *Gary*.

The engines are triple-expansion, with cylinders 24, 39 and 65 inches diameters by 42 inches stroke, supplied with steam from two Scotch boilers, 15 feet 4½ inches diameter by 11 feet 6 inches long, fitted with Ellis & Eaves induced draft and

allowed 170 pounds pressure. It is interesting to note that this is precisely the engine equipment of the *Manola*, which was built by Pickands, Mather & Co. in 1890, and which was at the time the largest carrier on the lakes. The *Manola* would carry in a full season 68,000 tons of ore. The *Corey* will carry on an equivalent number of trips over 225,000 tons of ore with the same fuel consumption. It is clear from these figures that the economical carrier is the single steamer of large carrying capacity and low power. The day of the consort is past.—Extract from "Marine Review."

The Fore River Shipbuilding Company, Quincy, Mass., closed a contract with the Southern Pacific Railway for a freight and passenger steamer for the Morgan line, to be fitted with twin screws and Curtis turbines. The boilers will be supplied by the Babcock & Wilcox Company. The principal dimensions of the new steamer will be as follows: Length over all, 440 feet 6 inches; length between perpendiculars, 415 feet 8 inches; beam, molded, 53 feet; depth molded to upper deck at side, 37 feet; depth molded to main deck at side, 29 feet; deck molded to lower deck at side, 20 feet 3 inches; height between saloon deck and upper deck, 7 feet 9 inches; height between promenade deck and saloon, 7 feet 6 inches; height between saloon deck and boat deck, 7 feet 6 inches; length of the amidship superstructure, about 182 feet; displacement, loaded, about 10,000 tons.—"Marine Review."

Turbine Steamers.—The W. & A. Fletcher Company, of Hoboken, N. J., has recently secured the contract for three turbine steamers. One of these, with 5,000 horsepower motors, is for the Eastern Steamship Company, and is intended for service between Boston, Mass., and Halifax, N. S. The others, of 10,000 horsepower each, are for the Metropolitan Steamship Company; they will run from New York to Boston.

Onward.—This turbine steamer, built by Messrs. Wm. Denny & Bros., Dumbarton, to the order of the Southeastern and Chatham Railway Co., proceeded to sea on her official trial trip. On the mile a mean speed of 22.54 knots was ob-

tained, a record speed for passenger turbine steamers, whilst with the turbines reversed there was a backward speed of 15½ knots. This latter feature, combined with the increased ahead speed, will mean a considerable saving of time in the channel passage. The official trial trip of the vessel consisted of a continuous run of six hours, when everything worked satisfactorily.

Invicta.—This turbine steamer, built by Messrs. Denny Bros., Dumbarton, to the order of the Southeastern and Chatham Railway Co., proceeded to sea on her official trial trip recently. On the measured mile at Skelmorlie her mean speed for a double run was 22.935 knots, or practically 23 knots. The contract condition was that one of the boilers should be out of action—that is to say, five-sixths of the boiler power was to be available—and under these conditions the speed attained was 21.85 knots. During this trial, too, the electric machinery, which is driven by a small steam turbine, was working at test pressure. As the guarantee speed for the four-hours' run was 21 knots, this trial also proved most satisfactory to the builders. The *Invicta's* turbines and auxiliary machinery, as well as the boilers, were constructed and fitted on board by Messrs. Denny & Co.

OBITUARY.

CAPTAIN THEODORE F. BURGDORFF, U. S. NAVY (RETIRED).

It is with regret that we announce the death of Captain Theodore F. Burgdorff, U. S. Navy (Retired), a member of this Society, at the U. S. Naval Hospital, Philadelphia, Pa., on July 29, 1905.

Captain Burgdorff was born on November 22, 1854, at Newark, N. J. He was appointed a Cadet Engineer ta the U. S. Naval Academy in 1873, was graduated in 1875, and on July 1, 1877, was commissioned an Assistant Engineer in the Navy.

On June 22, 1886, he was promoted to Passed Assistant Engineer, and to Chief Engineer on February 26, 1897. Upon the passage of the Naval Personnel Bill, March 3, 1899, his rank was changed to that of Lieutenant Commander, and on September 30, 1904, he was promoted to Commander.

Shortly before his death Captain Burgdorff made application for voluntary retirement, and on June 30th was placed on the retired list of the Navy with the rank of Captain.

His first cruise was made on the *Swatara*, at the expiration of which he was detailed for duty in the Bureau of Steam Engineering; after a cruise on the *Nipsic*, he was again ordered to the same Bureau.

Captain Burgdorff afterwards served on board the *Iroquois*, *Thetis*, *Oregon* and *Monadnock;* in 1903 and 1904 he served as Fleet Engineer of the South Atlantic Station on board the flagship *Newark.*

He was at various times detailed for duty on shore at the New York and Mare Island Navy Yards, and was for several years the Instructor of Engineering at the University of Tennessee.

Upon the completion of his cruise on the *Newark*, in October, 1904, he was ordered to inspection duty under the Bureau of Steam Engineering, and remained upon this duty up to the time of his death.

Captain Burgdorff ranked high in his profession, and in his death the Society and the Naval service have suffered a distinct loss.

BOOKS RECEIVED.

THE NAVAL POCKET BOOK FOR 1905.—By SIR W. LAIRD CLOWES, London; W. THOCHER & Co., 2 Creed Lane, E. C. Price, 7*s.* 6*d.* net.

This little book contains, in convenient form, much valuable information concerning the navies of the world.

Having been before the public for ten years, there is little need for further recommendation.

The following is a summary of the contents, the information given being brought up to date: Calendar and Naval Remembrances, 1905, Classification of Ships, Comparative Summary of the Fighting Fleets, Classified List of the Navies of the World, Guns and Small Arms, Notes on Torpedoes, Naval Personnel and Comparative Rank, Retiring Ages in European Navies, List and Particulars of Dry Docks, Miscellaneous Tables, Trial-Trip Tables, Plans of Ships.